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MAKING FRESH WATER BY DISTILLING SEA WATER IN 1792. EXPERIMENTS OF CAPTAIN JOHN FOSTER WILLIAMS, U. S. R. C. S.

BY REAR ADMIRAL ELLIOT SNOW (C. C.), U. S. N.

Reference: The American Museum or *Universal Magazine,* 1792 Part 1. Jany. to June. Princeton University Library 0901-164.

At a meeting of the Boston Marine Society held on the first of May, 1792,* John Foster Williams, Commander of the U. S. Revenue Cutter *Massachusetts,* communicated sundry experiments made by himself for the purpose of extracting fresh water from salt. This communication was accompanied with a plan of the apparatus† which he used while on a recent cruise also with

* From record attested by John Molineux, Secretary, addressed to Captain Mungo Mackay, President.

† A fruitless attempt to locate this sketch in the files of the Boston Marine Society was made early in the year 1928.

samples of water produced with it, which samples were found to be pure and were highly approved.

This record places the Revenue Cutter Service (U. S. Coast Guard) in the fore rank of having the credit for devising and using distilling apparatus at sea.

Upon the conclusion of the sitting a vote of thanks was given to Captain Williams, and the Society directed that his communication be printed in the *Columbian Centinel* and *Massachusetts Magazine* for the advantage of "our sea faring brethren."

It ran thus:

FRESH WATER FROM THE SEA

BY

JOHN FOSTER WILLIAMS

"To the President and Members of the Marine Society:

I enclose you the result of several experiments for the purpose of extracting fresh water from salt, made by me during the cruise of the Revenue Cutter.

You will observe that the apparatus made use of are such as are generally on board vessels at sea. I have also sent you samples of the water thus extracted, which I hope will not only meet the approbation of the Society, but prove of advantage to our sea faring brethren, in general, who may be so unfortunate as to be short of water when at sea.

I am, sirs, your humble servant,

Boston, JOHN FOSTER WILLIAMS."
May, 1792.

The first experiment to distill fresh water from salt conducted on March 6, 1792, is described as follows:

"I put four quarts of salt water in a sauce pan in the cabin stove; in fifty-five minutes I got from it near two quarts of good fresh water; one quart of water left in the sauce pan, the rest was lost. The *machine* made use of was a tin crane* with a barrel or cooler made to it of the same (material?) containing about three quarts with a hole in the top and bottom to put the cold water in.

* A siphon or bent pipe used for drawing liquid from a cask.

I let it out occasionally as it heats. I found that the barrel was not large enough to keep the tube cold. I then put five gallons of salt water in an iron pot — made the pot lid tight by putting some old canvas round it — made a hole in the middle (of the cover) with a hollow plug to receive the crane. I got from it two quarts of good fresh water in one hour and a half; but finding that my cooler was not large enough to keep the crane cool I left off for the time."

* * * * *

"April 11th — put in an iron pot 27 quarts of salt water and got from it

Of good fresh water	22 quarts
Left in pot	1 quart
Lost in seven hours	4 quarts
Total	27 quarts

The *machine** made use of here is as follows, viz., I took a hand pump and sawed it angle-wise and when put together it formed a crane; a gun barrel ran through a half barrel tub of water, with the end of it fixed to the crane — the pot-lid fixed with old canvas tacked around it, and made to fit very tight and secured down to prevent its rising; and shifted the water in the tub occasionally as it grew warm — *the cooler it is kept the better it will run.*

* * * * *

April 21. Made a *wooden tube* to use in the room of (in place of) the gun barrel through the tub of water. I put in *one quart* of *beans* in the pot with the twelve quarts of fresh water — in about one hour I got from it one quart of fresh water — it tasted very little of beans; but I found that wood did not answer so well as the gun barrel, *it being so thick that the cold water had not power over it to keep it cool.*

* * * * *

April 23. I cast a leaden tube and put it through the tub, put in the *machine* twenty quarts of salt water — got from it, in four hours, nine quarts of good fresh water: The pot was dry and

* Italics have been supplied; they do not appear in the original.

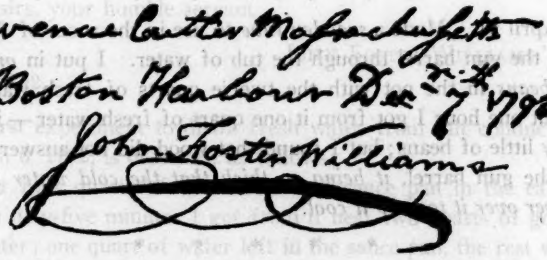
about a pint of the last running had a disagreeable taste. I took out of the pot about a pint of dry salt. I found the lead (pipe) was so thick I could not keep it cool; which was the occasion of my not getting more fresh water, as it went off in steam. I believe *if the lead was made very thin* it would answer very well.

* * * * *

April 24. Put in the pot eleven or twelve pounds of salt beef and sixteen quarts of salt water and in one hour and a quarter I got five quarts of good fresh water — it had a little taste of the beef and a very small appearance of grease on the top, which by filtering through linen cloth was taken off. With the above I made use of the gun barrel and find that in boiling anything at sea a considerable quantity of fresh water may be saved without expending any more wood than would answer to boil the meat, by having a hole through the pot-lid with a plug to it, to supply the water in the pot occasionally as it boils away.

In the iron tea-kettle that holds when full, five quarts, I put four quarts of salt water and fixed a wooden lid that was tight; made a hole through it to receive the crane; stopt the nose (of the tea-kettle) and with my *tin* crane in the cabin stove, in one hour and a half five pints of good fresh water (were made) and (nearly) three pints (were) left in the kettle."

JOHN FOSTER WILLIAMS.



The U. S. Revenue Cutter on which these experiments were carried out had the following dimensions and characteristics:*

* Taken from an inventory dated at Newport, R. I., April 10, 1816.

Length, fifty-eight feet six and one-half inches.
Breadth, seventeen feet nine inches.
Depth, seven feet.
Tonnage, sixty-two and forty-eight ninety-fifths.

She had two masts, one bowsprit, one flying jib boom, one main boom and gaff, and one fore boom and gaff.

THE PROPULSION CIRCUITS OF THE U. S. S. *LEXINGTON*.

By Lieutenant Edgar Kranzfelder, U. S. Navy, Member.

The fundamental principles of the system of propulsion installed on the U. S. S. *Lexington* and *Saratoga* are essentially similar to those of the U. S. S. *New Mexico* and later electric drive battleship installations. There are, however, numerous very interesting differences in the details of the propulsion system of the *Lexington* which will be particularly stressed during the course of this paper. These differences are principally due first, to the relative magnitudes of the power developed (the ratio of the maximum power developed by the *Lexington* is to that of an electric drive battleship, approximately, as six to one). Second, new developments in the electrical industry have had an important influence on the design of the propulsion system of the *Lexington*.

In order to get a comprehensive picture of the general scheme of the entire installation a survey of the calculated operating data in Table 1 will prove useful. To supply alternating current power to the eight 22,500 H.P. induction type propelling motors, four 40,000 KV.A. adjustable frequency turbo-alternators were installed. The system of control provides for seventeen different "set-ups" or combinations of motor and alternator connections to suit the various speed and power requirements of the vessel. There are two motors arranged in tandem on each propelling shaft; either or both motors can be energized depending upon the power requirements.

From the table of calculated operating data we see, for example, that it requires 57,000 shaft horespower to drive the *Lexington* at a speed of 25 knots. It will be noted that either 2 or 4 turbo-alternators and 4 or 8 motors can be used at that speed. In either case the speed of the alternators is 1255 R.P.M. Since it is characteristic of an induction motor to run within about 1 or 2 per

OPERATING DATA

CALCULATED VALUES BASED ON PROPELLER DATA

Knots	10	16	18	18	25	25	30	33.7
Propeller R.P.M.	91	144	162	162	226	226	274	317
Syn. Speed Motor	92.5	145.5	163.2	164	228	228	276	319
Generator R.P.M.	1020	1600	1800	900	1255	1255	1520	1755
Speed ratio	11:1	11:1	11:1	5.5:1	5.5:1	5.5:1	5.5:1	5.5:1
No. turbo generator	1	1	1	2	2	4	4	4
No. of motors	4	4	4	4	4	8	8	8
Total S.H.P.	4500	15000	22000	22000	57000	57000	101500	180000

MAIN MOTOR DATA

No. Poles	44	44	44	22	22	22	22	22
S.H.P. per motor	1125	3875	5500	5500	14250	7125	12685	22500
Torque per motor	65000	141000	179000	180000	333000	167000	242000	374000
Volts–line	1690	3880	4900	1790	3360	2380	3490	4980
Ampères per motor line	536	785	878	1610	2190	1550	1875	2310
Slip	.016	.010	.0085	.012	.009	.009	.0074	.006
P. F. per cent	60	60	60	88	88	88	88	88
Efficiency inc. ventilation	89.5	91.5	92	94	95	95	95.3	95.5

MAIN GENERATOR DATA

KV. A. per generator	6280	21100	29800	10000	25400	12800	22600	40000
Kw. per generator	3780	12650	17900	8800	22400	11250	19900	35200
Ampères–line	2150	3140	3515	3220	4380	3100	3750	4620
Frequency	34	53.3	60	30	41.8	41.8	50.7	58.5
Field ampères	398	601	668	465	643	473	571	694
Field volts	103	155	168	120	166	122	147	179
Total loss, inc. windage	284	708	923	308	568	414	658	961
Eff. guaranteed	91.8	93.8	94.4	96	97	95.8	96.2	97

MAIN TURBINE DATA

Steam press. gauge	265	265	265	265	265	265	265	265
Superheat F.	50	50	50	50	50	50	50	50
Vacuum (inches H. G.)	28.5	28.5	28.5	28.5	28.5	28.5	28.5	28.0
Guaranteed water rate	18	12.9	11.5	....	11 5	....	10.8	11.2

cent of synchronous speed we would expect the shaft R.P.M. to be within 1 or 2 per cent of some fixed multiple of the alternator speed. If then the alternators have 4 poles, as on the *Lexington,* and are running at 1255 R.P.M. and driving the 22-pole loaded induction motors the speed of the latter will be about 1 per cent slower than 4/22 or 1/5.5 of the alternator speed or 226 R.P.M. The synchronous speed of the motors under the above conditions is 228 R.P.M. The slip then is 2 R.P.M. or .9 per cent. If now with the same alternator speed of 1255 R.P.M. we arrange the motors to operate with the 44-pole connection, the propeller speed will be about 1 per cent slower than 4/44 or 1/11 of the alternator speed or 113 R.P.M. The propeller speed then depends upon the speed of the alternator and is always in the ratio (disregarding slip) of 1:5.5 or 1:11 depending on whether the 22- or 44-pole combination is in use.

The 44-pole combination may be used up to 18 knots or 162 propeller R.P.M. At this propeller speed the alternator is running at about 1800 R.P.M. When speeds above 18 knots are required the motor stator windings must be arranged for the 22-pole connection. With this connection, propeller speeds of 162 to 317 R.P.M. may be obtained by changing the alternator speed from about 900 R.P.M. to about 1755 R.P.M.

Since the induction motor is inherently approximately a constant speed motor, several special provisions are necessary to make the motor suitable for marine use where both speed and torque vary over a wide range. The variation in speed is obtained, as has been pointed out above, by changing the frequency of the voltage impressed on the stator of the motor. For a constant voltage the synchronous speed of an induction motor varies directly as the frequency. As the speed of the motors is varied it becomes necessary to vary the impressed voltage so that the motors will operate at a good power factor and still "keep in step." To obtain a high torque at starting, it is necessary to momentarily impress a voltage on the motors considerably in excess of the normal operating voltage. The starting torque of an induction motor is greatest when the resistance of the rotor equals the rotor reactance. Under these conditions the starting torque is directly proportional to the square of the voltage impressed on the stator.

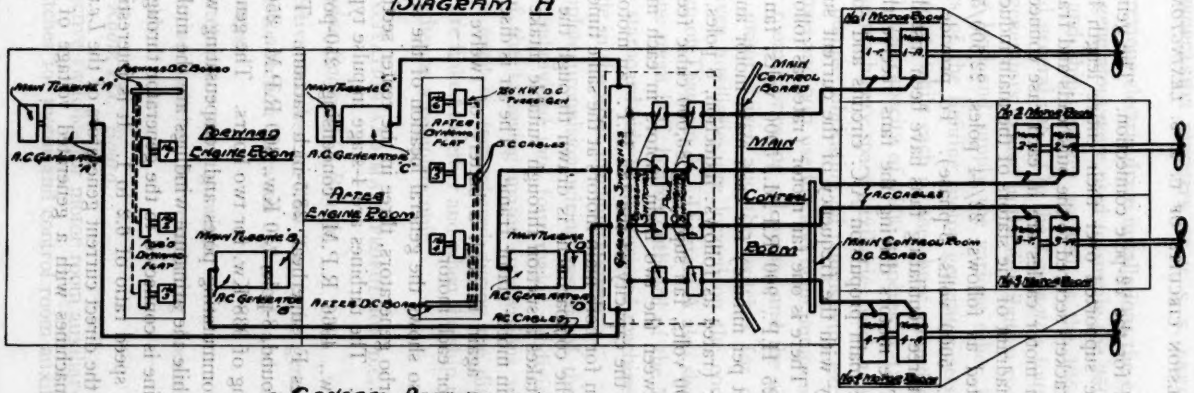
The rotors of the *Lexington's* motors have two windings, a high resistance squirrel cage winding which is effective during starting, and a low resistance form wound winding which is the running winding. The squirrel cage winding, with the resistance more nearly equal to the reactance of the winding, gives the high torque for starting while the low resistance winding with lower copper loss, less heating, and consequently greater efficiency, is effective after the motor is "in step."

If now the load on the motor increases, the motor slows down causing the rotor to slip more rapidly through the revolving field and resulting in an increase in the rotor current. But at the same time the numerical value of the rotor current increases, the frequency of that current is also increasing. Since the rotor reactance is a function of the frequency the reactance increases, which in turn increases the angle of lag of the current. Also, since the torque developed by the rotor increases due to the increase in the rotor current and the effect of the increase in the current lag is to decrease the torque, a certain point can be reached where the effect of the increase in the current lag is greater than the increase in the rotor current. At this point, called the pull out or breakdown point, the motor falls "out of step" because the torque decreases even though the motor current is increasing. Since for the same slip, the torque varies as the square of the voltage, the necessity for a very flexible control of the voltage impressed on marine propulsion induction motors can be appreciated. For the voltage must be varied, not only in starting when the impressed voltage must momentarily be considerably higher than normal voltage, but also under running conditions where the torque demands on the motors vary, as in making turns, maneuvering, or operating in shallow waters. Since the voltage generated by the alternator is nearly proportional to the value of the field current of the alternator, this offers a very satisfactory method of controlling the voltage applied to the motor stators. The field circuit of the alternator has been called "the heart of the control" because the entire system of electric propulsion centers about the alternator field circuits. All switching such as changing from one of the 17 "set ups" to another, is accomplished with the generator

field circuits open. The mechanical, electrical and pneumatically operated interlocks are so arranged that none of the above mentioned changes can be made except when the field circuits are open.

With these preliminary remarks in mind let us now turn our attention to Diagram A which shows the general arrangement of the main units and cables. It will be noted that the generator ends of the A.C. turbo generators in each space face each other and that the cables from the forward alternators run along the port side to the main control room while those from the alternators in the after machinery space run along the starboard side to the control room. The turbo alternators (each rated as follows: *turbine* — 13 stage, 35,200 Kw., 1755 R.P.M., 265-pound gauge steam, 50 degree F. superheat, 28.5 vacuum; *alternator* — 3-phase, revolving field, 40,000 KV.A., 1755 R.P.M., 5000 volt) are designated from starboard to port and forward to aft as: Generator A, Generator B, Generator C, and Generator D. Between each main turbo generator and the generator switch group in the main control high tension cell there are twelve 2,000,000 C.M. single conductor cables, *i. e.*, four cables per phase. The high tension cell, shown in broken lines, is located in the forward part of main control room. The control gear in the cell consists essentially of four high voltage switch groups arranged fore and aft and secured to a special foundation built into the ship's structure. Each group consists of a set of air break contactor type generator switches, a set of reversing contactors, and a set of pole changing contactors. It will be noted that no oil switches are used in the propulsion circuits of the *Lexington.* The control groups mentioned above are mounted and secured together in such a way as to serve as a self-contained support for the busbars without direct attachment to any of the bulkheads. The incoming cables from the generators are connected to the busbars which are mounted in two layers directly above the control groups. The busbars are clamped to upright insulated rods, which are supported from the control groups below. The outgoing cables to the propelling motors and fan motors are led out from the lower part of the control groups and run aft under the control room deck through the wiring passage to the motor rooms. There are nine 2,000,000 C.M. cables per motor, one per phase for the 44-pole connection and 2 per

DIAGRAM A



~ GENERAL ARRANGEMENT OF MAIN PROPULSION UNITS AND CABLES ~

phase in parallel for the 22-pole connection. The generator and motor cables are supported over their entire length by bakelite insulators and brackets secured to the bulkheads and frames of the ship. The main motor cables are in each case connected to the lower inboard quadrant of the stators of the main induction motor (each motor rated as follows: 22/44 poles, 22,500/5500 H.P., 317/162 R.P.M., 5000 volts, 3-phase). To provide ventilation for the motors, three ventilating fans have been installed in each motor room. The motors driving the fans receive their power supply from the main propulsion A.C. circuits and consequently their speeds vary with the frequency of the current supply from the generators. There is one fan motor (rated as follows: induction, 8 poles, 125 H.P., 900 R.P.M., 5000 volts, fan supplying 42,500 cubic feet per minute) for each main motor and an additional fan motor (rated as follows: induction, 8 poles, 250 H.P., 900 R.P.M., 5000 volts, fan supplying 85,000 cubic feet per minute) located between the two main motors in each motor room which has twice the capacity of the 125 H.P. fan motors and can supply ventilation for both main motors at the same time. In each case the air in the compartment is drawn through the motors by the fans, which takes a suction through suitable intake ducts connected to the main motor stator frame. The air is discharged into the motor room again through a battery of twelve surface air cooler sections for each motor.

Diagram A also shows the general location of the six 750 Kw. direct current turbo generators, three in the center section of each machinery space. The turbines are 4-stage impulse type rated as follows: 750 Kw., 4966 R.P.M., condensing, 250-pound gauge steam, 50 degrees F. superheat, 28.5-inch vacuum. The generators are shunt wound, 8-pole, 750 Kw., 800 R.P.M., 250-volt with an overload rating of 938 Kw. for two hours. The generators are equipped with commutating poles and compensating windings in the pole faces while the armature windings are the multiple drum type. The turbine is connected to the generator through a reduction gear with a speed ratio of 6.2 to 1. It is interesting to note at this time that the direct current generators of the *Lexington* are straight shunt machines with a generated voltage of 250 volts,

while on previous electric drive installations the compound wound three-wire generators were used to supply the 250-volt source for excitation and ship's power as well as the 125-volt source for lighting and general use. On the *Lexington* four 125 Kw. lighting motor generators located in the distribution rooms take care of the lighting load and supply the various other 125-volt circuits. The motors of the lighting motor generator sets operate on the 250-volt circuit, while each of the shunt wound generators has a normal full load capacity of 1000 ampères at 125 volts.

All of the machinery space auxiliaries, such as main circulators, condensate pumps, lubricating oil pumps, etc., except the port use fuel oil pumps and port use fire and bilge pumps are steam driven. The motor room lubricating oil pumps and the air and oil cooler pumps are motor driven and the switches for controlling these electric driven auxiliaries are located on the control room D.C. board shown on Diagram A. Diagram A also shows the locations of the engine room D.C. boards outboard of the 750 Kw. generators on the starboard side in the forward machinery space and on the port side in the after machinery space.

The complete propulsion wiring system of the *Lexington* can logically be subdivided into four major subdivisions. Although each one of the following major subdivisions will be discussed separately it must be remembered during the discussion that all the circuits are linked together and when the ship is underway they operate together as a complete unit.

A. EXCITATION AND MAIN PROPULSION D.C. CIRCUITS.

1. The Engine Room D.C. Boards.

(*a*) Selective switches for 750 Kw. turbo generator sets.

(*b*) Interconnection between forward and after machinery spaces.

(*c*) Booster motor starting switch.

(*d*) Selective switches for connecting booster generators to fields.

2. Control Room D.C. Board.

(*a*) Selective switches for motor room auxiliary supply.

(*b*) Switches for motor and control room auxiliaries.

(*c*) Control bus selective switches.
(*d*) Selective switches for connecting booster sets or emergency rheostats to main generator fields.

3. Booster Motor Generators.

(*a*) Booster field connections and control.
(*b*) Main field contactors and connections.

4. Emergency Rheostat.

B. MAIN A.C. WIRING.

1. Power Wiring Between Generators and Motors.

(*a*) Generator contactors.
(*b*) Reversing contactors.
(*c*) Fan motor contactors.
(*d*) Pole changing contactors.

2. Main Motor Rotor and Stator Wiring Diagrams.

(*a*) Some fundamental points of the theory involved.
(*b*) The 44-pole connection.
(*c*) The 22-pole connection.

C. PNEUMATIC CIRCUITS.

(*a*) Sources of air supply.
(*b*) Reversing air engine.
(*c*) Hydraulic timer.
(*d*) Pole changing engine.
(*e*) Generator switch engine.
(*f*) Air valves and circuits of pneumatic control.
(*g*) Mechanical interlocks.

D. PROTECTIVE CIRCUITS, TESTING CIRCUITS, CONTROL CIRCUITS, WIRING FOR METERS.

(*a*) Protective relays.
(*b*) Testing circuits for protective relays.
(*c*) Phase failure relays.
(*d*) Interlocking circuits.

(*e*) Wiring for meters and recording instruments.
(*f*) Temperature indicator wiring.
(*g*) Speed and steam limit control circuits.

The sketches used in conjunction with the explanations of the wiring circuits are mostly simplified line sketches. In a few instances the diagrams used are portions of complete diagrams of the wiring circuits.

A. EXCITATION AND MAIN PROPULSION D.C. CIRCUITS.

1. The Engine Room D.C. Switchboards.

Each one of the D.C. engine room switchboards shown in Diagram A controls three 750 Kw., 240-volt, 2-wire, D.C., shunt wound, turbo-generators; two booster motor generator sets of 60 Kw. capacity each with a 40- to 200-volt generator directly connected to the 50 H.P., 240-volt D.C. motor. In addition there are switches to the shore connections, to the distribution switchboards, and D.C. supply for the main control room D.C. board.

Let us now turn our attention to Diagram B which shows the connections to the forward machinery space D.C. switchboard. The D.C. switchboard in the after machinery space is practically identical to the forward D.C. board except that the shore connection switch on the forward switchboard is double throw while on the switchboard in the after machinery space it is single throw. The leads from the shore connection boxes on the second deck connect to the center legs of the two S.P.D.T. switches on the forward machinery space D.C. board. Throwing these two S.P.D.T. switches marked (10) to the right energizes the forward engine room D.C. board while throwing the switches to the left energizes the after engine room D.C. board.

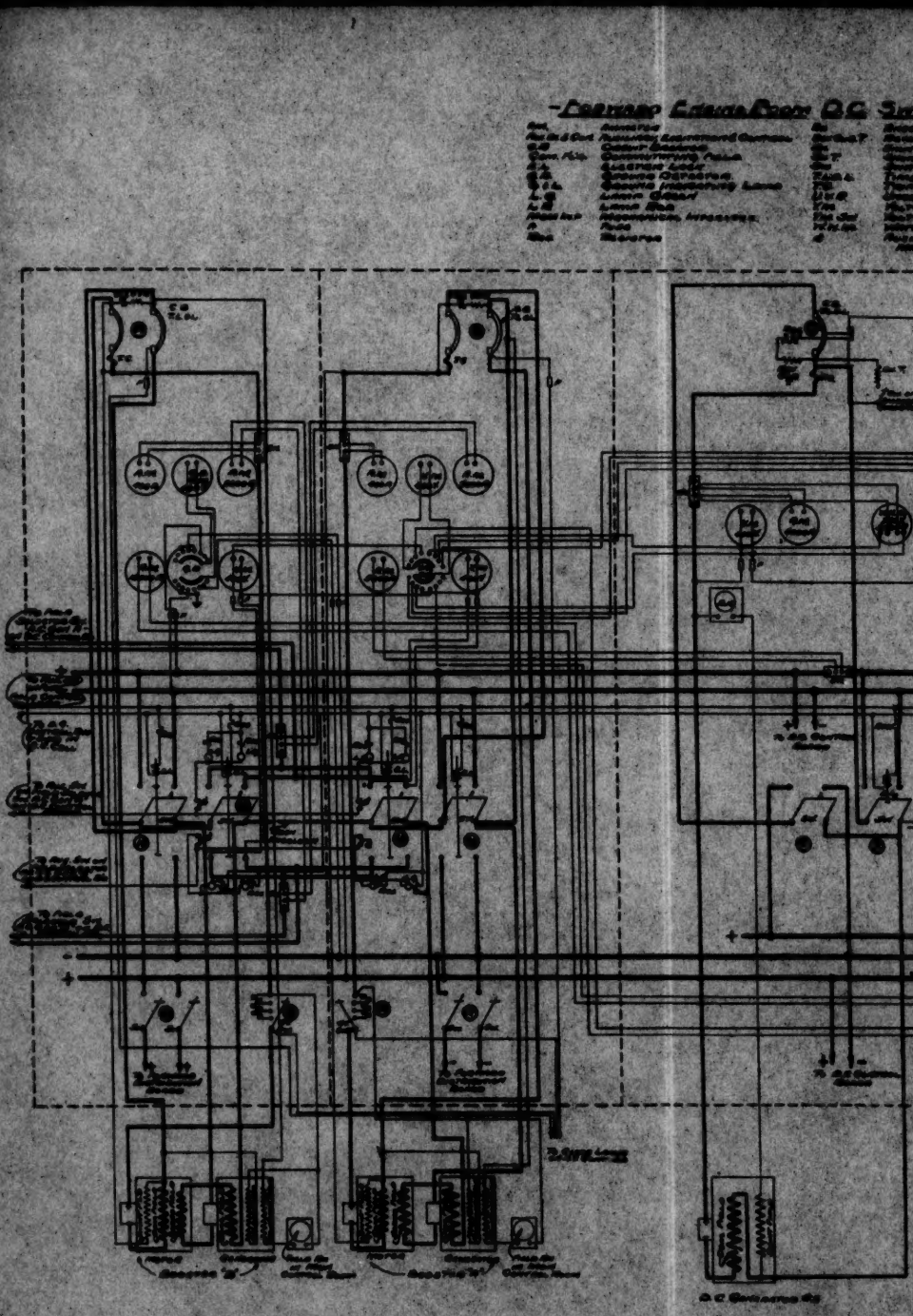
(*a*) Selective switches for 750 Kw. turbo generator sets.

It will be noted from Diagram B that each 750 Kw. turbo generator may be connected to either the ship's service bus or to the auxiliary and excitation bus or to both at the same time. Closing switches 2, 4 and 6 would connect the respective 750 Kw. generators to the ship's service bus which feeds the ship's light and power circuits, while closing switches 1, 3 and 5 would connect the generators to the auxiliary and excitation bus. It will be

further noted that each 750 Kw. generator is provided with one single pole, automatic, time limit, overload, shunt trip and reverse current circuit breaker marked 7, 8 and 9. The total amount of current delivered by the generator, that is, for ship's light and power, excitation, and main drive auxiliaries passes through the overload coil of the circuit breaker but only that part of the current used for ship's light and power passes through the circuit breaker contacts. The connections to the left-hand blades of switches 1, 3 and 5 feeding the excitation bus are tapped off on the generator side of the circuit breakers between the overload coil and the breaker contacts. With this arrangement if one of the 750 Kw. generators become overloaded due to a short circuit or an overload of the lighting and power circuits, the breaker will trip out but the supply to the auxiliary and excitation bus will not be interrupted.

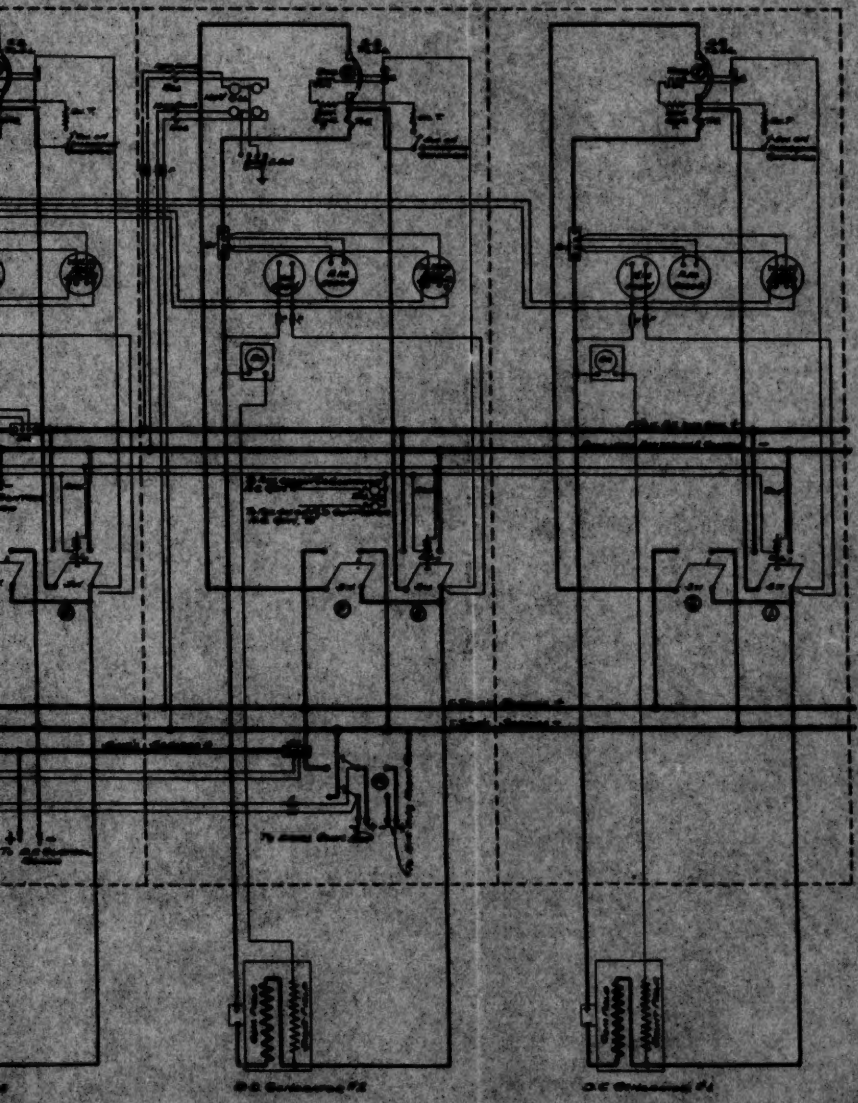
When two 750 Kw. generators in one machinery space are used for excitation and ship's service the two machines are usually operated in parallel on the ship service bus while the auxiliary and excitation switch is closed on one of the two generators. It can readily be understood that if two auxiliary switches were closed and one of the turbines tripped out the generator of the set which tripped out would probably be seriously damaged.

The handles of the auxiliary switches 1, 3 and 5 are painted red and a mechanical latch is provided on each switch as a reminder to the operator not to open any one of the switches without thinking nor to parallel any two generators for excitation. The latch may be released if so desired for temporary paralleling on the auxiliary bus while changing from one machine to another during maneuvers. To prevent the engine room D.C. switchboard operator from opening an excitation switch while the main field lever is in the "on" position electric interlocks are provided which are energized from the field control levers in the main control room. The interlocks are gravity closed and they only open when the field control levers are in the "off" position which energizes the interlocks through auxiliary switches on the generator field control levers. The circuit breakers 7, 8 and 9 are provided with a shunt trip shown in Diagram B, the purpose of which is to open the breaker when the emergency governor trips the turbine. Likewise



D.C. Switchgear -

DIAGRAM B



a reverse current trip is provided which opens the breaker when the current reverses when one of two machines operating in parallel trips out. Diagram B also shows the small auxiliary switch marked (*a*) which opens when the circuit breaker opens and thereby de-energizes the reverse current trip and the shunt trip.

(*b*) Interconnection between the forward and after machinery spaces.

It is well to note at this time that the only way the forward and after engine room D.C. switchboards can be paralleled is through the distribution switchboards. Because the shore connection switches are S.P.D.T. on the forward engine room D.C. board and S.P.S.T. on the after engine room board these two switchboards cannot be paralleled through the shore connection switches. When switches 17 and 18, Diagram B, are closed the forward distribution board can be energized and likewise when the corresponding switches on the after engine room D.C. switchboard are closed the after distribution board can be energized. The two distribution boards can then be paralleled through circuit breakers and bus tie switches.

(*c*) Booster motor starting switch.

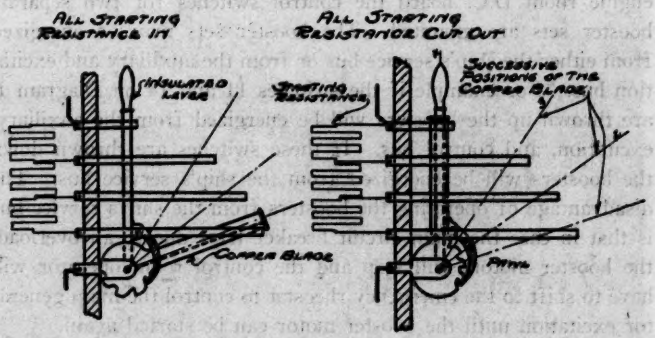
On the right-hand side (facing the front of board) of each engine room D.C. board the control switches for two separate booster sets are mounted. The booster sets may be energized from either the ship's service bus or from the auxiliary and excitation bus. For example if the switches 11 and 14 on Diagram B are thrown up the boosters will be energized from the auxiliary, excitation, and control bus. If these switches are thrown down the boosters will be energized from the ship's service bus. The disadvantage of operating the boosters from the ship's service bus is that in case the main circuit breaker trips due to an overload, the booster motors will stop and the control room operator will have to shift to the emergency rheostat to control the main generator excitation until the booster motor can be started again.

Energizing the booster from the auxiliary and excitation bus does not present this difficulty because, as pointed out in a previous paragraph, this bus remains energized as long as the main

auxiliary switch is in, even though the 750 Kw. circuit breaker has tripped.

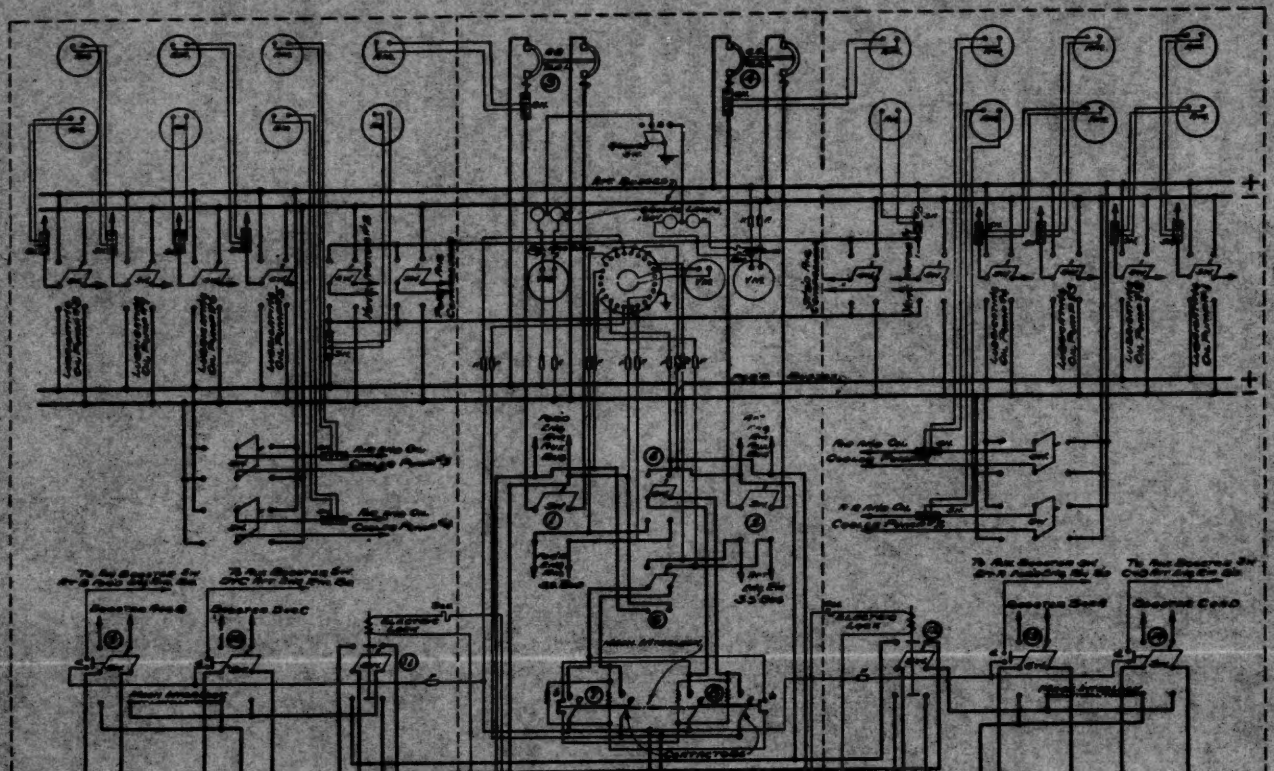
The booster motors are started with a special pump type starting switch marked 15 or 16 on Diagram B. The circuit for booster A, for example, comes from either the ship's service or auxiliary bus from the switch 11, through the double pole, time limit, overload circuit breaker 19. The starting switch 16 is located in the positive lead to the booster motor armature. The detail Diagram C of the "pump" type starting switch shows that it consists of an insulated arm, a copper blade, and a pawl arrangement. During the first throw of the switch the insulated arm drags the copper blade after it by means of the pawl. This operation connects the booster motor armature to the source of power through the resistances shown in Diagram C. The pumping operation is continued and at each step more resistance is cut out until finally when the copper blade is in the vertical position the entire resistance is short circuited. This pumping action introduces a time element which allows the motor to come up to speed evenly as the operator cuts out the successive resistor units in the circuit.

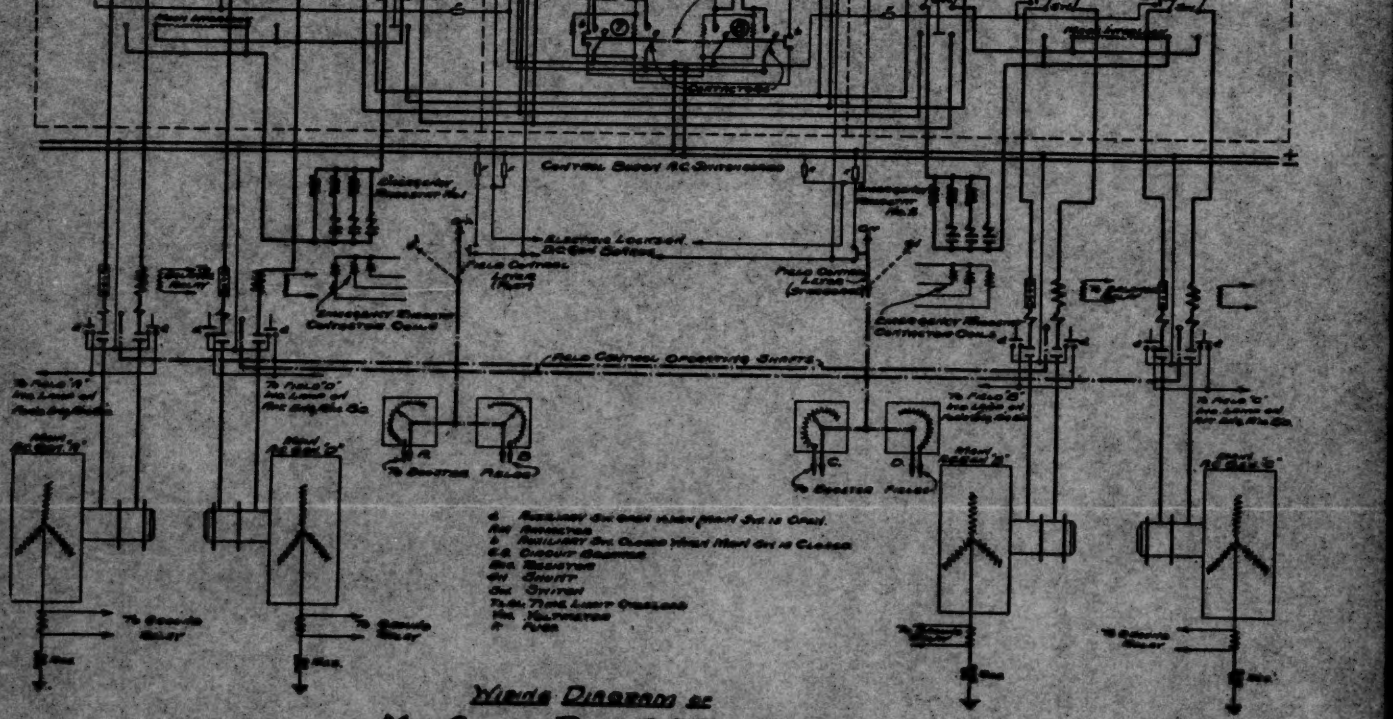
DIAGRAM C



"PUMP" STARTING SWITCH FOR THE BOOSTER MOTOR

E.H.





Wiring Diagram of
— Main Control Room D.C. Switchboard —
Diagram D

(*d*) Selective switches for connecting booster generator to main generator fields.

To provide greater flexibility in the use of the booster sets the D.P.D.T. switches 12 and 13 on Diagram B were installed. For example, by throwing switch 12 up booster A is used on generator A field provided the selector switch on the main control room D.C. board is thrown up. If switch 12 is placed in the lower position, booster A is connected to the field of generator B provided, of course, the selector switch on the main control room D.C. board is thrown up. A mechanical interlock between the two upper throws of the switches 12 and 13 and between the two lower positions of these switches is provided so that the two boosters cannot be placed in parallel on one field circuit.

In order that the field supply switches on the booster control will not be opened after once closed unless the main field control lever is in the off position, electric interlocks shown on Diagram B are provided. These interlocks are similar to the magnetic interlocks on the auxiliary switches connecting the 750 Kw. generators to the auxiliary excitation and control bus. The red lights at the selector switches 12 and 13 show normal operation, *i. e.*, when booster A is used with generator A, etc., while the green lights show when, for example, booster A is used with generator B.

2. Control Room D.C. Board.

(*a*) Selective switches for motor room auxiliary supply.

Diagram D shows the detail wiring of the main control room D.C. switchboard which is located on the upper platform in the after end of the main control room. Other than the supply to the booster selective switches there are four sources of current supply for the control room D.C. board, *i. e.*, from the forward machinery space auxiliary and excitation bus, from the forward machinery space ship's service bus, from the after machinery space auxiliary and excitation bus, or from the after machinery space ship's service bus. Which source of power is to be used for the motor and control room auxiliaries depends, first of all, upon the position of the main double pole double throw, selective switches Nos. 1 and 2. The center or hinge clips of switch 1 are connected through

the double pole time limit, overload circuit breaker 3 to the forward bus which feeds the various auxiliaries. Thus when switch 1 is in the up position the forward bus of the control room D.C. board is being energized from the forward machinery space auxiliary and excitation bus, while putting switch 1 in the down position energizes the forward bus of the main control room D.C. board from the forward machinery space ship's service bus. Similarly by means of switch 2 the after bus of the control room D.C. board may be energized from the after machinery space auxiliary and excitation bus or from the ship's service bus.

(*b*) Switches for motor and control room auxiliaries.

The individual double pole, double throw, unfused lever switches are mounted on the upper portion of the main control room D.C. board. There is a separate switch for each one of the following auxiliaries: 8 motor room lubricating oil pumps, 4 motor room air and oil cooler pumps, 2 main control room ventilating motors and the 2 main control room railway type air compressors which supply air to the control room pneumatic system. It will be noted that when these double pole double throw switches are up, or to the right in the case of the air and oil cooler pumps, the current is being supplied from the after buses, either auxiliary and excitation or ship's service depending upon the position of the main selective switch No. 1. Similarly when the double pole double throw switches are down, or to the left in the case of the air and oil cooler pumps, the current is being supplied from the forward buses, either auxiliary and excitation or ship's service depending upon the position of the main selective switch No. 2. Each one of the above mentioned individual switches has an ammeter in the positive outgoing lead, except the two ventilating motors and the two air compressor motors, which are totalized on one meter for forward bus and one for the after bus.

(*c*) The control selective switches.

A similar arrangement to the switches 1 and 2 is provided in the case of the control circuit which is energized through either switch 5 or switch 6. Depending upon the positions of switches 5 and 6 the control may be energized from: (1) forward engine room auxiliary excitation and control bus when switch 6 is down, (2) forward engine room ship's service bus when switch 5 is

down, (3) after engine room auxiliary and excitation bus when switch 5 is up, (4) after engine room ship's service bus when switch 6 is up.

In order that only one source of supply can feed the control bus, *i. e.*, in order that two sources of supply on the control bus cannot be paralleled, the supply to the control bus passes through two double pole contactors, 7 and 8. These contactors are mechanically interlocked so that only one set can be in at one time. Let us say, for example, that the control bus is being supplied from the forward machinery space auxiliary excitation and control bus (switch 6 down) and that switch 5 is up. If now the forward auxiliary bus supply fails, the contactors 7 fall out by gravity. As they fall out the auxiliary contact b on 7 closes energizing the holding coils of contactors 8 and closing these contactors. This puts the control bus supply on the after engine auxiliary excitation and control circuit through switch No. 5. The control bus supplies current for electric interlocks, the temperature indicating circuits, the speed and steam limit motors and all the indicating lights on the main control board and on the D.C. boards.

(*d*) Selecting switches for connecting booster sets or emergency rheostats to main generator fields.

Referring again to Diagram D we find that the double pole, double throw switches 11 and 12 serve to energize the emergency rheostats. The two emergency rheostats Nos. 1 and 2 are so connected to the control room D.C. board that No. 1 rheostat can be used on either the field of A alternator or of D alternator and No. 2 rheostat can be used on either the field of B alternator or of C alternator. The purpose of this arrangement is evident since with only two rheostats available the greatest possible flexibility in the use of these rheostats is necessary. Switches 9 and 10 when in the up position connect the booster sets to the main generator fields A and D respectively. Likewise when switches 13 and 14 are in the up position the booster sets are connected to the fields of B and C generators respectively. When switches 9, 10 or 14 are in the lower throws they connect the emergency rheostats to the generator fields. However, switches 9 and 10 are mechanically interlocked so that emergency rheostat No. 1 can only be used with generator A or D. With this locking

arrangement only one emergency rheostat can be used on one main generator field at one time. Similarly emergency rheostat No. 2 can only be used with generator fields B or C. When switches 9, 10, 13 and 14 are in the up position an auxiliary contact on each switch is closed which in each case completes the circuits back to the indicating lights on the selective switches 12 and 13 on Diagram B of the machinery space D.C. boards. These lights show the operator on the D.C. machinery space switch board when the selective switches are closed and warn him not to open his switches while the lights are burning.

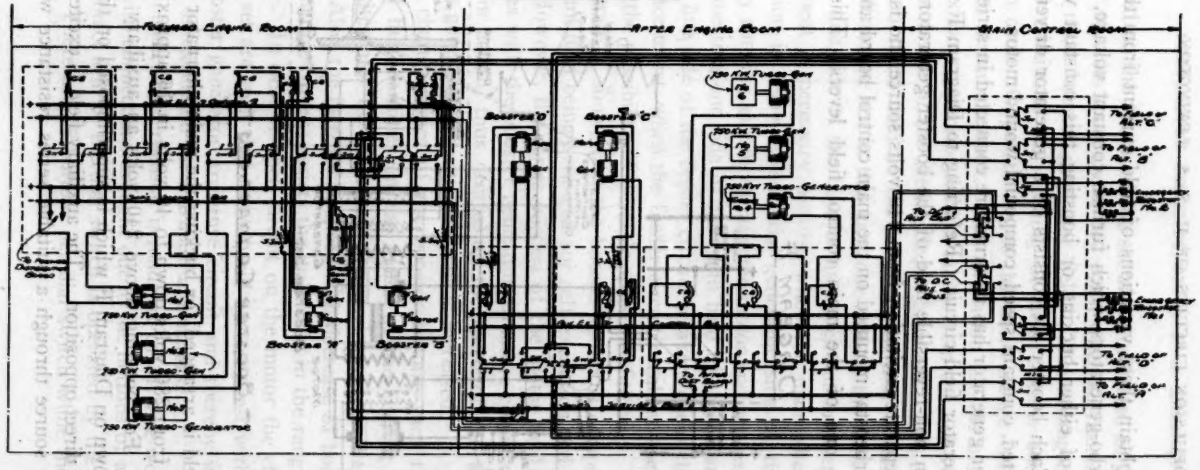
Both the machinery space switchboards and the main control room D.C. board are equipped with the necessary ammeters, voltmeters, ground detecting lamps, shunts, instrument fuses.

With the general wiring scheme of the machinery space D.C. switchboards and the main control room D.C. board in mind let us glance at Diagram E. This diagram shows in general the interconnections between the three D.C. boards. It shows first, how booster A can be used with alternator field on A or B and how booster set C can be used with alternator field C or D. Similar for booster sets B and D. Diagram E also shows how emergency rheostat No. 1 can be used with the field of A or D but not with B or C, how emergency rheostat No. 2 can be used with the field of alternator A or D. Diagram E further shows the shore connection and that the forward and after machinery space D.C. board cannot be paralleled through the shore connection because of the double throw switches on the forward switchboard and the single throw switches on the after machinery space switchboard, the shore connection leads being secured to the center legs of the forward machinery space S.P.D.T. switches.

3. Booster Motor Generators.

Let us refer back for a minute to the calculated operating data on page 547. We find for example that at 10 knots a field current of about 398 ampères and a voltage of about 103 volts is required, while at 18 knots the current required is about 668 ampères with a field voltage of about 168. Also, as has been mentioned before, different conditions of the load and sea and also in starting or reversing, require very different values of field current.

Diagram E.

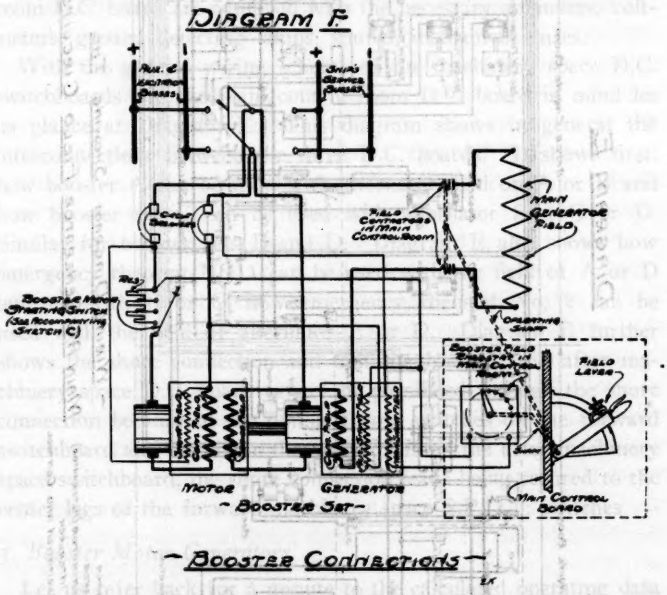


SW. Switch
S.S.W. Starting Switch
C.B. Circuit Breaker
Alt. Main A.C. Generator (Alternator)
M.I. Mechanical Interlock
Aux Exc'n Auxiliary Excitation and Control

~ Wiring Diagram Showing The Main AC Generator Field Circuits ~

In order to obtain these variations of field current from the 750 Kw. D.C. turbo-generators which furnish constant voltage, boosters, capable of either bucking or boosting this constant voltage are used. Each booster set consists of a generator driven by a direct connected, constant speed, compound wound motor.

The booster generator has its armature connected in series with the main generator field circuit. Referring to Diagram F we see that the main non-reversible field of the booster generator is arranged for separate excitation from 240 volts source and is under control of a rheostat mounted on the main control board and controlled from one of the main control field levers. This non-



BOOSTER CONNECTIONS

reversible field is arranged for bucking the main alternator excitation voltage from 280 volts down to 40 volts in 76 approximately equal steps. For boosting above 240 volts an auxiliary field is provided shown on Diagram F which is superimposed on the main fields and in direct opposition. The auxiliary field is excited from the 240-volt source through a suitable series resistance which is

mounted on the frame of the set and is adjusted to give approximately 40 volts boost above 240 volts, when all resistance is cut in on the main booster generator field. This arrangement of field control eliminates the opening of field circuits and provides a smooth cycle of operation over the entire range from maximum boost to maximum buck. The excitation for the motor field and the auxiliary generator field is taken from the 240-volt source on the motor side of the feeder circuit breakers. The excitation for the main shunt field of the booster generator is taken from the 240-volt source but in this case one side of the field is connected on the line side of the booster motor circuit breakers. This arrangement is necessary because when the main alternation field excitation is below normal the generator side of the booster is actually running as a motor and the motor end acts as a generator. So connecting one side of the main field of the booster generator on the line side of the feeder circuit breakers opens this field, with reference to set, when the feeder circuit breakers open thus eliminating the possibility of unstable speed conditions.

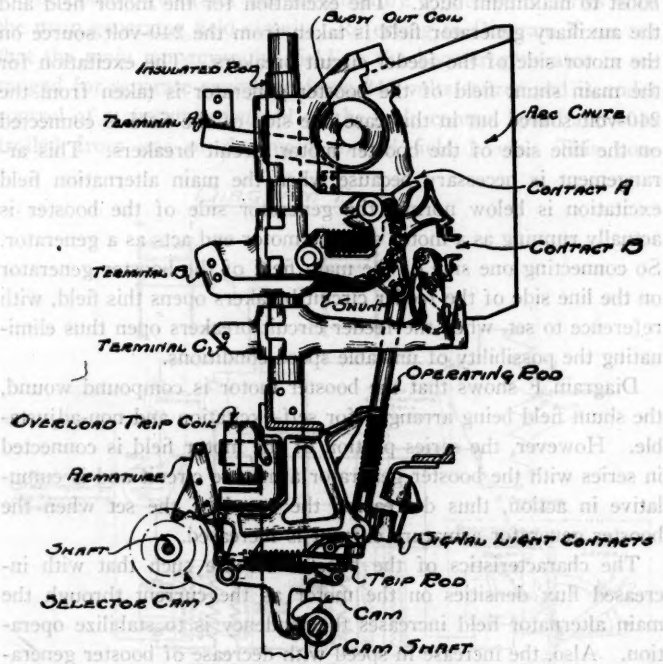
Diagram F shows that the booster motor is compound wound, the shunt field being arranged for self-excitation and non-adjustable. However, the series portion of the motor field is connected in series with the booster generator armature circuit and is cumulative in action, thus decreasing the speed of the set when the booster generator armature current is increased.

The characteristics of the booster set are such that with increased flux densities on the motor as the current through the main alternator field increases the tendency is to stabilize operation. Also, the increase in speed with decrease of booster generator armature current assists the main field in the range of bucking voltage. In case of open circuit on the motor the characteristics of the set are such that the set will come to rest without opening the booster generator armature circuit and thereby insuring continuity of excitation on the main alternators.

The booster field rheostat is made up of 31 wire-wound resistor units with connections brought out to the buttons of a dial switch. The contact arm of the dial switch is operated by the field control lever. There are 78 buttons per dial switch while the total resis-

tance of the rheostat is about 256 ohms varying from .6 to 35 ohms per step. The current through the main booster field varies from 0.3 to 4.5 amperes.

DIAGRAM G



CROSS SECTION THROUGH A FIELD CONTACTOR SHOWING THE OPERATING MECHANISM. E.K.

(*b*) Main field contactor and connections.

It is interesting at this time to examine Diagram G which is a sketch of a field contactor. There are two field contactors per alternator, one in each side of the line to the main alternator field.

Each contactor is rated to carry 800 ampères continuously at 500 volts D.C. with a temperature rise not to exceed 50 degrees C.

The current conducting parts of the contactor are secured to upright steel rods covered with insulation of great electrical and mechanical strength and considerable flexibility. The three terminals A, B and C are clamped to the insulated rod to provide for the single pole, double throw connection required in order to place a discharge resistance across the alternator field whenever the field is opened. When the main contact A is closed the current enters from the supply at the terminal A and passes from there into the series blow out coil, from the series blow out coil to the upper contact finger, through the flexible shunt shown in the diagram, to the outgoing terminal B which is connected to the generator field. When contact A is closed the auxiliary contact B is open, but as soon as contact A opens contact B closes and the discharge resistance is connected across the alternator field as shown in Diagram Y. The circuit through the field contactor is from the field, to terminal B, through the large flexible shunt to upper tip of contact B, to lower tip of contact B through the small flexible shunt, to the support to which terminal C is attached, through the discharge resistance to the other side of the field. Both contacts A and B are surrounded by an asbestos compound arc chute and both sets of contactors come within the effect of the magnetic field developed by the series blowout coil.

The construction of the operating mechanism of the field contactor is also very interesting. The main contact A is closed and opened by means of the operating cam shaft which is moved manually by the field control lever on the front of the control panel. When the cam is rotated it raises the cam lever and the trip rod which normally engages the operating rod and closes the contact A. When the cam lever is lowered contact A is opened and contact B closes by force of gravity and through the force of the operating rod spring. It will be noted from Diagram G that the trip rod slides between a set of rollers which form a part of the cam lever. The trip rod is attached to the armature which is moved either by the selection cam or when the trip coil is energized. This pulls the trip rod to the left from under the operating

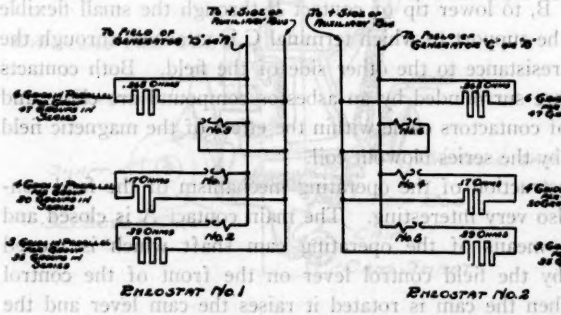
rod and the contact A opens and contact B closes. When one of the two units in a machinery space is not in use the selector cam, which is connected to the main selector shaft, cuts out the field contactors for that unit by keeping the trip rod to the left so that when the cam rotates the trip rod will not engage the operating shaft.

Auxiliary contacts are mounted on the operating rod which light up signal lights on the main control board and show when the contact A is closed.

4. The Emergency Rheostat.

In case of trouble with booster sets two separate emergency rheostats are provided. As has been previously explained, one of these emergency rheostats may be used with either alternator field A or D while the other emergency rheostats may be used with either alternator field B or C. Diagram H shows a simplified con-

DIAGRAM H



WIRING DIAGRAM OF EMERGENCY RHEOSTATS.

nection diagram of the grids of the rheostat and the contactors. Each rheostat is provided with resistance steps so connected that when the alternator field is connected to the 240-volt excitation bus, the following values of field current can be obtained:

1st, with all contactors open, the first section made up of six grids in parallel is in series with the alternator field. This section

of resistance amounts to approximately .235 ohm and gives approximately 500 ampères in the alternator field which corresponds to minimum excitation current.

2nd, the section made up of groups of three grids in parallel is connected in parallel with the first section, when contactor No. 2 is closed. This gives approximately .35 ohm resistance and 600 ampères which corresponds to the average load running current.

3rd, the section made up of four grids in parallel is in effect when contactor No. 1 is closed. The grids in this section have a total resistance of .15 ohm and are placed in parallel with the other two sections giving approximately 700 ampères in the alternator field which corresponds to the maximum load running current.

4th, when contactor No. 3 is closed all of the resistance is short circuited and the alternator field is connected directly across the 240-volt bars. This gives approximately 900 ampères in the field and corresponds to the maximum "pull in" current.

The various contactors of the emergency rheostats are closed electrically by two small levers on the main control board. In operating the emergency rheostat the main field control lever must be used in conjunction with the emergency rheostat so that the motor rotor switches will open and close making the high resistance squirrel cage winding effective during starting.

B. MAIN A.C. WIRING.

1. Power Wiring Between Generators and Motors.

Let us now turn our attention to the alternating current part of the installation. The source of the A.C. is of course the alternators, which because of their likeness to shore alternators will be merely described as 4-pole, 3-phase, revolving field machines with a rating of 40,000 KV.A. at 1755 R.P.M. and 5000 volts.

Diagram J shows the A.C. wiring between the alternators and the main propelling motors. It will be noted that current supply passes first, through the generator switches, then through the reversing switches and finally, through the pole changing switches. By referring to the supplementary Diagram I the method of connecting the four alternators to the eight propelling motors in the

17 different selections is evident. If, for example, it is desired to use generators A and C, selection No. 6, then the generator switches Nos. 1, 4, 7 and 9 would be closed and switches 2, 3, 5, 6, 8 and 10 would remain open. With this arrangement or "set up" generator A would supply motors 1F or 1A and 2F or 2A,

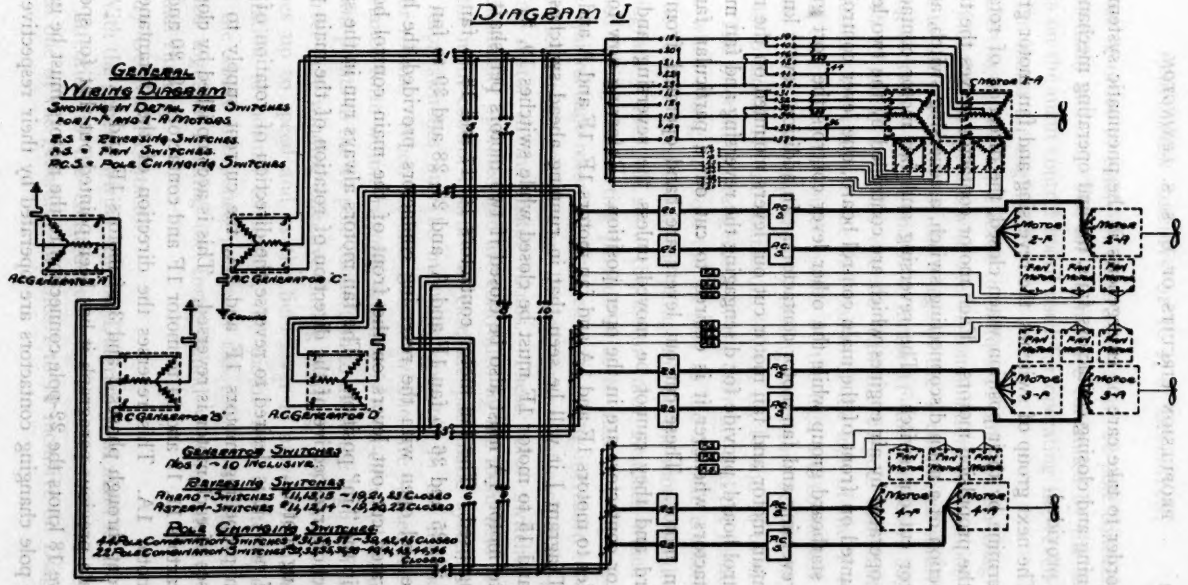
DIAGRAM I

~ GENERATOR SELECTOR TABLE ~ SHOWING THE GENERATOR SWITCHES WHICH ARE CLOSED FOR THE VARIOUS SET UPS

SELECTION	SPEEDS	GENERATORS IN USE				GENERATOR SWITCHES TO BE CLOSED			
1	FULL	A	B	D	C	4	3	2	1
2	UP TO 26 KNOTS	A		B		5	4	7	9
3		B		D		2	3	7	9
4		A		D		2	4	7	9
5		B		C		1	3	7	9
6		A		C		1	4	7	9
7		D		C		1	6	7	9
8	UP TO 19 KNOTS	A				4	8	9	10
9		B				3	5	8	9
10		D				2	6	7	9
11		C				1	7	8	10
12	SPECIAL EMERGENCY CONDITION	A	D		C	1	2	4	8
13		D	B		C	1	3	6	8
14		A	B		C	1	3	4	8
15		A	D		B	2	4	5	8
16		C	B	D	C	1	2	3	10
17		A	B	D	A	2	3	4	10
		1-A 4-F	3-A 3-F	2-A 2-F	1-A 1-F				
		PORT MOTORS		STARBOARD MOTORS					

while generator C would supply motors 3F or 3A and 4F or 4A. These generator switches are closed and opened by means of cam shafts which extend the length of each control group. The cam shafts are normally operated by means of the generator air engines.

DIAGRAM J


General
Wiring Diagram
Showing in Detail the Switches for 1-A and 1-A Motors.
R.S. = Reversing Switches.
P.S. = Part Switches.
P.C.S. = Pole Changing Switches
A.C. Generator 'A'
A.C. Generator 'B'
A.C. Generator 'C'
A.C. Generator 'D'
Ground
Motor 1-A
Motor 2-A
Motor 3-A
Motor 4-A
Part Motor
P.C. 1
P.C. 2
P.C. 3
P.C. 4
Generator Switches
Nos 1 ~ 10 Inclusive.
Reversing Switches
Ahead - Switches 11,12,13 ~ 19,21,23 Closed
Astern - Switches 14,15,16 ~ 17,20,22 Closed
Pole Changing Switches
44 Pole Combination Switches 31,34,37 ~ 30,42,45 Closed
22 Pole Combination Switches 32,33,35,36,38 ~ 40,41,43,44,46 Closed

In order to take care of emergencies if the pneumatic system for opening and closing contactors fails, manual operating mechanisms are provided.

The next group of switches, the reversing and fan motor group, determine by their position when closed, the direction of rotation of the propulsion motors. Each motor contactor serves the triple function of a main disconnecting switch, a reversing switch, and a motor cut out switch. The reversing and fan motor contactors are operated by air engines which are controlled by the two levers mounted on front of the main control board, one lever controlling the starboard group while the other lever controls the port group of reversing and fan motor contactors. In addition, levers known as main motor and fan motor cut out levers mounted on the main control board, provide for disengaging the reversing and fan motor contactors whenever it is desired to cut out a particular fan or main motor. These cut out levers are hand operated from the board and they cannot be moved unless the reversing and fan motor contactors are in the open position. Referring now to the cables to motors 1F and 1A and fan motors 1F, 1F and A and 1A, on Diagram J it will be seen that in running ahead switches 11, 13 and 15 to motor 1F must be closed while switches 19, 21 and 23 to motor 1A must also be closed if two motors per shaft are being used. The fan motor contactors 16, 17 and 18 to fan 1F, and 24, 25 and 26 to fan 1F and A, and 27, 28 and 29 to fan 1A, close and open with the reversing contactors provided the hand operated cut out levers on the front of the main control board are in the "in" position. The fan motors always run in the same direction irrespective of the direction of rotation of the main propelling motors.

If now, it is desired to reverse the direction of rotation of the main propelling motors 1F and 1A, the current supply to two phases of each motor is reversed. This is accomplished by closing contactors 11, 12 and 14 to motor 1F and contactors 19, 20 and 22 to motor 1A. This reverses the direction of the instantaneous current through phases 2 and 3 of motors 1F and 1A.

In a previous paragraph it has been pointed out that for speeds above 18 knots the 22-pole connection to the motors must be used. The pole changing contactors are operated by their respective air

engines in order to establish either the 44- or 22-pole connection of the main motors. The air engines are controlled by the same levers on the main control board used for operating the reversing and fan motor air engines.

Referring again to the leads to motors 1F and 1A on Diagram J we find that to operate with the 44 combination, connections are made to the outer ends of the 3 legs of the Y connected stator by closing contactors 31, 34 and 37 to motor 1F and contactors 39, 42 and 45 to motor 1A. To operate with the 22-pole combination connections are made to the mid points of the legs of the Y connected stator and the outer points of the legs are connected together. This is accomplished by closing contactors 32, 33, 35, 36 and 38 to motor 1F and contactors 40, 41, 43, 44 and 46 to motor 1A. The internal rearrangement of the currents and the resultant poles when changing from the 44-pole to the 22-pole connection will be further discussed in the paragraph on the motor windings.

Diagram J does not show the motor rotor switches which are opened during the starting periods so as to make the high resistance squirrel cage rotor windings effective giving increased torque. Two motor rotor switches are located forward of each main propelling motor in the motor rooms. These rotor switches are pneumatically operated and are controlled by the main field levers on the main control board. In case the pneumatic system fails the rotor switches may be manually operated in the individual motor rooms. When the motor rotor switches are opened the low resistance form wound running winding is open-circuited between certain points in the 44-pole connection to make the high resistance squirrel cage winding effective during starting. When the motor comes up to speed and as the field lever is moved from super excitation to normal excitation, the motor rotor switches close and the above mentioned points in the rotor winding are connected together, making the form wound winding effective for normal running.

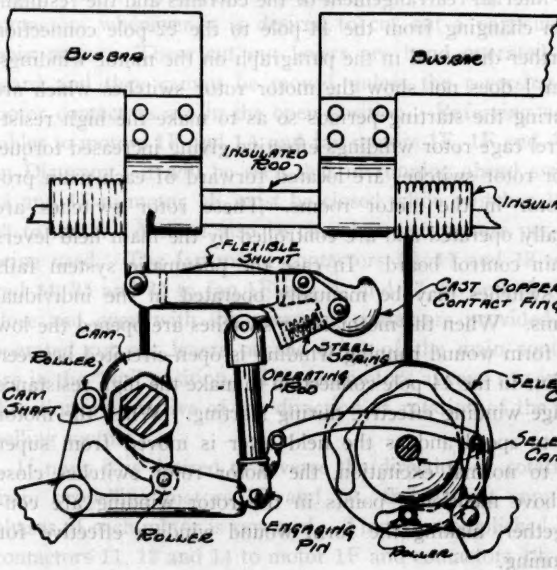
(*a*) Generator contactor.

With the general wiring scheme, Diagram J, in mind let us turn our attention to the construction of the individual contactors. Each one of the generator switches 1 to 10 inclusive on Diagram

J is composed of three generator contactors, each serving as a disconnecting switch in one phase of the three-phase circuits.

The rating of each generator contactor is approximately 4650 ampères at 5000 volts, 60 cycles with a temperature rise not to exceed 50 degrees C. Diagram K is a cross section through one of the contactors showing how the contact making parts are mounted on a single insulated horizontal rod. The rod is of cold rolled steel covered with wrappings of laminated fibre and mica giving a construction of great electrical and mechanical strength with considerable flexibility. Contact is made through a multiple set of cast copper fingers which are maintained in contact by

DIAGRAM K



CROSS SECTION THROUGH A GENERATOR CONTACTOR (GENERATOR SWITCH)

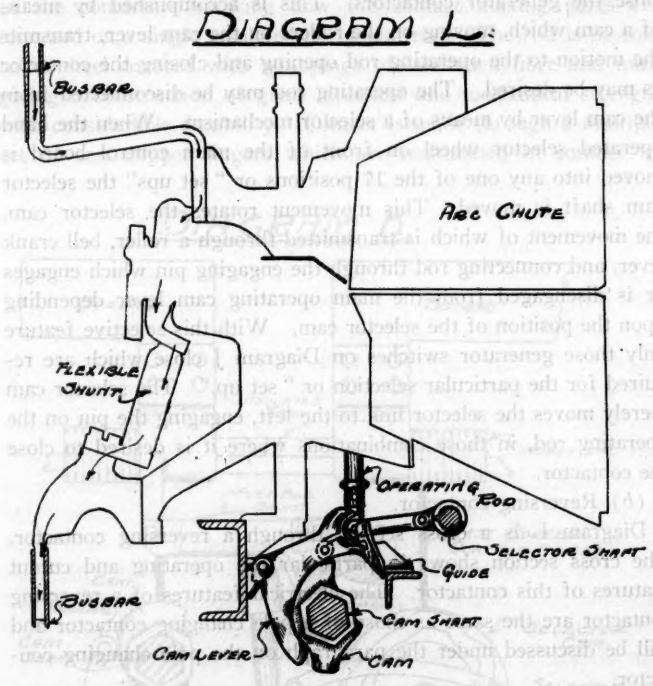
powerful steel springs. The current passing from one terminal to the other is shunted around the above-mentioned steel spring and also around the pivot bearing of the switch by means of the flexible shunts. The operating cam shaft shown on Diagram K is rotated by the generator air engine and serves to open and close the generator contactors. This is accomplished by means of a cam which, moving on the rollers on the cam lever, transmits the motion to the operating rod opening and closing the contactor as may be desired. The operating rod may be disconnected from the cam lever by means of a selector mechanism. When the hand operated selector wheel on front of the main control board is moved into any one of the 17 positions or "set ups" the selector cam shaft is moved. This movement rotates the selector cam, the movement of which is transmitted through a roller, bell crank lever, and connecting rod through the engaging pin which engages or is disengaged from the main operating cam lever depending upon the position of the selector cam. With this selective feature only those generator switches on Diagram J close which are required for the particular selection or "set up." The selector cam merely moves the selector link to the left, engaging the pin on the operating rod, in those combinations where it is desired to close the contactor.

(*b*) Reversing contactor.

Diagram L is a cross section through a reversing contactor. The cross section shows in particular the operating and cutout features of this contactor. The electrical features of a reversing contactor are the same as those of a pole changing contactor and will be discussed under the paragraph on the pole changing contactor.

A cam and a cam lever connect the operating rod to the main operating shaft. As in the case of the generator contactor, when the operating shaft is rotated by the reversing air engines the action of the cam closes and opens the switch, provided the operating rod is engaged with the slot of the cam lever as shown in Diagram L. An additional mechanism is provided for disengaging the operating rod from the cam lever so that when the cam shaft is rotated the cam lever will not engage the operating rod and in this way the contactor remains open. When the selec-

tor shaft, which is operated manually from the main control board is rotated, the link connected to the selector shaft moves to the right. The contactor must be in the open position in order to perform this operation. When the link moves to the right it pulls the operating rod out of engagement with the main operating cam



CROSS SECTION THROUGH A REVERSING CONTACTOR SHOWING THE OPERATING AND CUTOUT MECHANISM.

lever so that when the cam lever is moved by the air engine the contactor remains in the open position.

(*c*) Fan motor contactor.

Each fan motor contactor is rated to carry continuously 25 ampères at 5000 volts. Each fan motor contactor is equipped

with magnetic blowout coils, but the operating mechanism is interlocked so it cannot open the contactor until the field control lever on the main control board is moved to the "off" position and then only after a predetermined time has elapsed as provided by the hydraulic timer on the air engines.

The methods of supporting the contactor and the rod insulation are similar to the other type of contactors. The essential features of the contact making parts and the magnetic blowouts are the same as those of the pole changing contactor which will be discussed in the following paragraph. The operating mechanism and the motor cutout mechanism of the fan motor contactors are practically the same as for the reversing contactors described in a former paragraph.

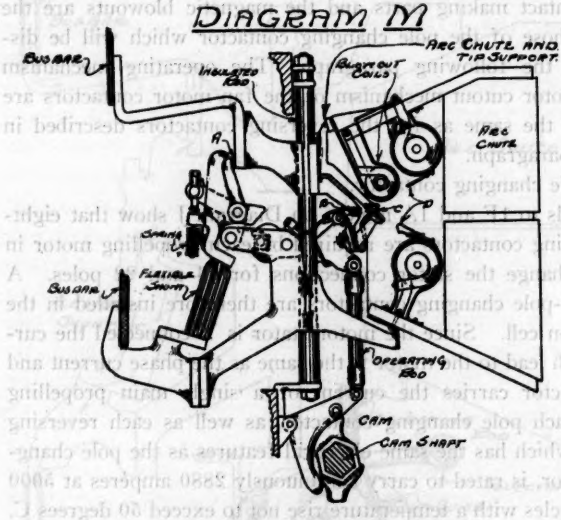
(*d*) Pole changing contactor.

The leads to 1F and 1A motors on Diagram J show that eight-pole changing contactors are required by each propelling motor in order to change the stator connections for 44 and 22 poles. A total of 64-pole changing contactors are therefore installed in the high tension cell. Since the motor stator is Y connected the current in each lead to the motor is the same as the phase current and each contactor carries the current of a single main propelling motor. Each pole changing contactor, as well as each reversing contactor which has the same electrical features as the pole changing contactor, is rated to carry continuously 2880 ampères at 5000 volts, 60 cycles with a temperature rise not to exceed 50 degrees C. The contactors are equipped with magnetic blowout coils, but, as in the case of the reversing and fan motor contactors, the operating mechanism is interlocked so it cannot open the contactors until the field control lever is moved to the "off" position, and then, only after a predetermined time has elapsed as provided by the hydraulic timer on the air engine.

The use of the magnetic blowout type of contactor on the *Lexington* in place of the oil switch marks one of the applications of new developments in the electrical industry to marine electric propulsion installations. The arc rupturing capacity of this type of magnetic blowout contactor was carefully tested at the Schenectady plant of the General Electric Company at 60 cycles, 3-phase

with a water box load. One contactor was connected into each of the three phases of a test alternator. The contactors were then opened and closed a number of times at various voltages and currents up to rated load, in each case with full field on the alternator.

Diagram M is a cross section through a pole changing contactor showing the method of support, the operating mechanism, the con-



tact making parts, and the magnetic blowout. The method of support is similar to that used for the reversing and fan motor contactors. The insulation of the upright steel rod to which the high voltage parts of the contactor are secured consists of a combination of thin sheets of fibre and mica which are coated with a binding compound and wrapped around the rod under heat and pressure. Over this is tightly wound two wrappings of cotton tape also saturated with the binding compound and the entire insulation is baked for several hours. With this construction the surfaces of

the supports are vertical, smooth and non-hygroscopic. These qualities prevent the collection of dust or other particles that might effect its insulating properties.

The operating mechanism of the pole changing contactor is practically the same as that of a reversing contactor except that in the case of the pole changing contactor no cutout mechanism is provided. The pole changing contactors are in line along the cam shaft and operate in two combinations only, *i. e.*, one for 44 poles and the other for 22 poles. There is no "off" position for the pole changing contactors. The system of using a cam shaft for operating the contactors, provides an excellent means for operating a large number of contactors simultaneously, as well as a simple and positive means of interlocking between units which insures the proper sequence of operation of the contactors.

There are a number of interesting points about the electrical features, *i. e.*, the contact making parts and the magnetic blowouts, of the pole changing, reversing and fan motor contactors. Referring again to Diagram M it will be noted that the main current carrying contacts located on the left-hand side of insulated supporting rod are lettered A. The arc rupturing contacts B and C as well as the arcing horns D are located on the right-hand side of the insulated rod in Diagram M. When the contactor is fully closed, the current at one instant enters the upper bus bar, flows through the upper main terminal, through the main contact finger at contact A, through the flexible shunt, through the terminal and out through the lower bus bar. When the contactor begins to open under load, the contacts A open first while arc rupturing contacts B and C remain closed. When contacts A are open the current flows through B, through the small flexible shunt from lower contact of B to the lower terminal support and out to the lower bus bar. Since contacts B and A are in parallel and since B contacts open after A contacts have opened, they insure that the arc will only be ruptured in the asbestos compound arc chute surrounding contacts B, C and the arcing horns D. As the contactor opens a little more the contacts of B open developing an arc at B of considerable resistance. The current, seeking the path of least resistance, now flows from the upper arc chute and contact tip support to one end of the flat copper strip of the blowout coil,

through the small series blowout coil, through the contact tip support, through contacts C, through the small flexible shunt to the lower terminal support and out through the lower bus bar. When the contactor opens far enough so C contacts open, the arc is maintained for an instant and then the magnetic blowout effect transfers the arc to the stationary arcing horn tips of D. When this happens two additional powerful magnetic blowout coils, shown in Diagram M, are added in the circuit. The current now flows from the arc chute and contact tip support through the small upper series blowout coil, through the large upper blowout coil to the upper arcing horn of D. From the upper arcing horn of D the current flows through the arc to lower arcing horn of D, through the lower blowout coil, through the lower terminal support and out through the lower bus bar. The powerful magnetic field developed by the blowout coils causes the arc to move from the contact tips to the arcing horns D. The arcing horns are shaped so as to stretch the arc and draw it away from the contact tips of B and C. As the arc is stretched the magnetic field is intensified by the additional blowout coils and the arc is moved rapidly to the rupturing point. It must be remembered that after the main alternator field has been opened and the reversing levers on the main control board placed in the reverse position, the hydraulic timer automatically introduces a delay of approximately 7 seconds before the contactors begin to open. This permits the current to be ruptured by the contactors to die down to a certain predetermined value.

2. *Main Motor Rotor and Stator Wiring Diagrams.*

Among the most interesting electric features of the installation on board the *Lexington* are the motor windings. All eight propelling motors are exact duplicates in so far as the electrical characteristics and capacity are concerned and differ only in details of mechanical construction to suit the installation requirements of the ship. As has been previously stated, each motor is wound three-phase and is arranged with connection suitable for establishing either a 44-pole or a 22-pole rotating field. The motor rotors are provided with a high resistance squirrel cage winding for securing high starting torque and a low resistance winding for operation

in step. The low resistance form wound winding is opened and closed between certain points in the winding in the 44-pole connection, by two pneumatically operated contactors controlled by the main field levers on the main control board. With the low resistance winding open between these points on the 44-pole combination, the high resistance squirrel cage winding is effective. There are three ways then, in which the electrical behavior of a main propelling motor might be altered:

1st, stator connected for 44-pole operation with the rings on the rotor open circuited, *i. e.*, with the low resistance winding open.

2nd, stator connected for 44-pole operation, with the rings on the rotor short circuited, *i. e.*, with the low resistance winding closed.

3rd, stator connected for 22-pole operation with the rings on the rotor closed.

The first condition with the 44-pole stator and the collector rings open, the only effective winding on the rotor is the high resistance squirrel cage in the bottom of the rotor slots, this winding giving a high torque near standstill. As will be pointed out in a following paragraph on the 44-pole connection, no current flows in the form wound winding.

The second condition is the normal operating condition for low speed. The rotor of the motor is now in effect a six-phase rotor when considering the action of the definite winding on the rotor in the 44-pole field and in order for currents to flow in the form wound rotor winding the collector rings must be short circuited.

The third condition is the normal high speed operating condition. It is immaterial whether the collector rings are open or short circuited since for the 22-pole stator connection the rotor is in effect connected in 12 phases and is internally short circuited. The rings are then at equipotential points in the rotor winding. The high resistance squirrel cage winding does not give high starting torque in the 22-pole connection because the form wound winding takes practically all of the induced current which is developed.

In order to show how 22 and 44 poles are established in the stator of the motor and the resultant effect of the revolving field on the motor rotor, it will be necessary to investigate the motor

windings to some extent. There are 396 slots in the stator of each motor and in these 396 slots there are placed 396 coils. Therefore in each slot there are two coil sides; that is, the right-hand side of one coil and the left-hand side of another. Facing the stator toward the end to which the incoming cable connections are made the right-hand coil side of all the coils are placed at the bottom of the slot and the left-hand coil sides of all coils are placed at the top. Each one of the 396 coils is made up of 8 turns in series of rectangular wire enameled and then covered with extra heavy cotton covering impregnated with varnish. Insulation is placed between turns in the slot part of the coil and the end of every other turn is taped to avoid short circuiting between turns. Each coil is then insulated with four layers of mica tape wound half lap, followed by windings of linen tape, then treated with moisture resisting varnish and finally thoroughly baked. With a total of 396 coils on the stator there are 132 coils for each phase of the 3-phase winding.

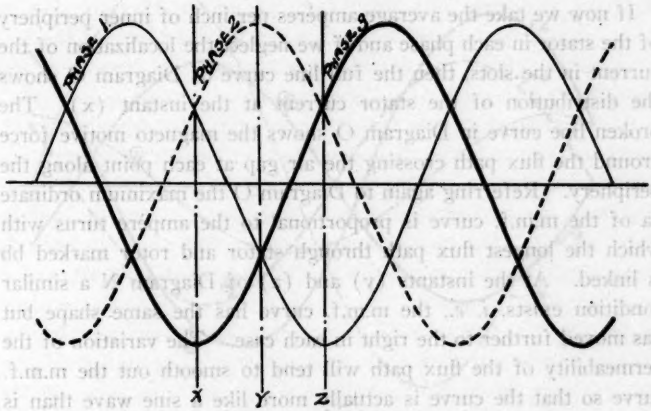
The rotor has 528 slots, with two independent windings in the same slots. In the bottom of the slots are the high resistance bars of the squirrel cage winding. In the top of the slots are assembled 1056 insulated bars of the phase winding, *i. e.*, two bars per slot.

(*a*) A few fundamental points of the theory involved.

Before continuing with the discussion of the motor windings a review of some of the fundamentals concerning the current and voltage relations in the stator and rotor should help to clarify the directions of the currents and locations of poles in the wiring Diagrams T and V.

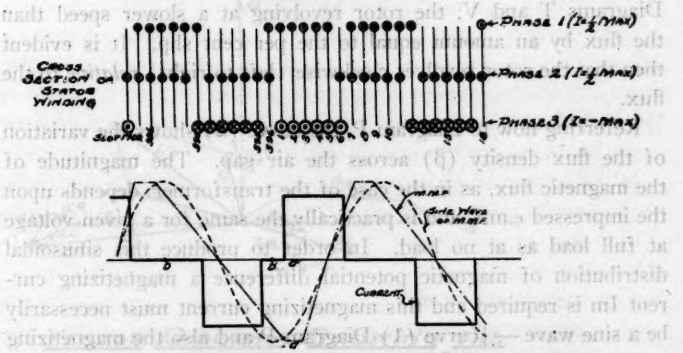
Let us start with Diagram N which shows the currents (sine wave assumed) which flow at any instant in the three phases of the stator, the current in phase 1 leading the current in phase 2 by 120 electrical degrees and similarly the current in phase 2 leading the current in phase 3 by 120 electrical degrees. The upper part of Diagram O shows a schematic cross section of the stator winding between slots 384 and 18 inclusive, the dots representing instantaneous currents directed out of the paper and the crosses representing instantaneous currents directed into the paper. If we select for example the instant (x) on Diagram N the current

DIAGRAM N



CURRENT CURVES FOR STATOR WINDING

DIAGRAM O



DISTRIBUTION OF STATOR CURRENT AND M.M.F. AT THE INSTANT (X) ON DIAGRAM N

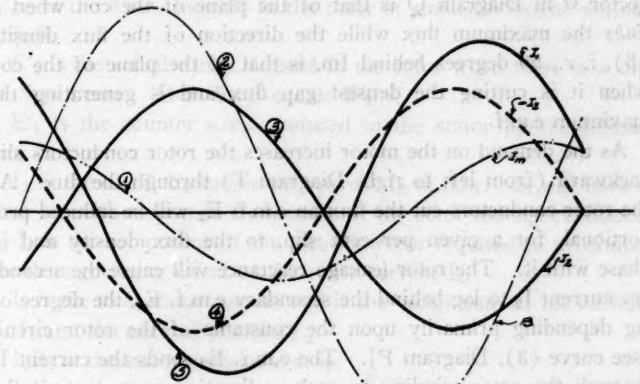
in phase 1 is one-half its maximum positive value, the current in phase 2 is also one-half its maximum positive value, while the current in phase 3 is at its maximum negative value.

If now we take the average ampères per inch of inner periphery of the stator in each phase and if we neglect the localization of the current in the slots, then the full line curve of Diagram O shows the distribution of the stator current at the instant (x). The broken line curve in Diagram O shows the magneto motive force around the flux path crossing the air gap at each point along the periphery. Referring again to Diagram O the maximum ordinate aa of the m.m.f. curve is proportional to the ampère turns with which the longest flux path through stator and rotor marked bb is linked. At the instants (y) and (z) of Diagram N a similar condition exists, *i. e.*, the m.m.f. curve has the same shape but has moved further to the right in each case. The variation of the permeability of the flux path will tend to smooth out the m.m.f. curve so that the curve is actually more like a sine wave than is shown on Diagram O.

In order to simplify the explanation of the vector relations let us assume that during the remaining discussion of the motor windings that we have a sinusoidal flux and current distribution around the gap periphery at any instant. Also it is assumed that the flux and the rotor are revolving counter clockwise (right to left) in Diagrams T and V, the rotor revolving at a slower speed than the flux by an amount equal to the per cent slip. It is evident then that the rotor revolves clockwise (left to right) *relative* to the flux.

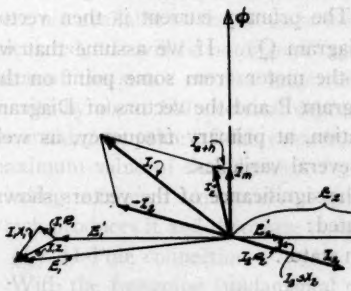
Referring now to Diagram P, the curve (2) shows the variation of the flux density (β) across the air gap. The magnitude of the magnetic flux, as in the case of the transformer, depends upon the impressed e.m.f. and is practically the same for a given voltage at full load as at no load. In order to produce this sinusoidal distribution of magnetic potential difference a magnetizing current Im is required and this magnetizing current must necessarily be a sine wave — [curve (1) Diagram P and also the magnetizing current vector Im in Diagram Q]. It will be noted that the mutual flux $\emptyset$ is in phase with the magnetizing current Im Diagram Q, and that in Diagram P the *flux density* (β) lags the

DIAGRAM P



CURVES SHOWING THE PERIPHERAL DISTRIBUTION OF FLUX AND CURRENTS.

DIAGRAM Q



VECTOR DIAGRAM SHOWING THE RELATIONS OF FLUX, VOLTAGE, AND CURRENT.

magnetizing current Im by 90 degrees. To clarify this point we must visualize the coil on the motor where the direction of the vector $\emptyset$ in Diagram Q is that of the plane of the coil when it *links* the maximum flux while the direction of the flux density (β), *i. e.*, 90 degrees behind Im, is that of the plane of the coil when it is cutting the densest gap flux and is generating the maximum e.m.f.

As the demand on the motor increases the rotor conductors slip backward (from left to right Diagram T) through the flux. As the rotor conductors cut the flux an e.m.f. E_2 will be induced proportional, for a given per cent slip, to the flux density and in phase with it. The rotor leakage reactance will cause the secondary current I_2 to lag behind the secondary e.m.f. E_2, the degree of lag depending primarily upon the constants of the rotor circuit [see curve (3), Diagram P]. The e.m.f. E_2 sends the current I_2 through the rotor winding in such a direction as to oppose the change of the flux $\emptyset$ which produces it and therefore to oppose the primary current I_1 [curve (5), Diagram P]. But the magnetizing effect of the primary current I_1 must always be greater than the demagnetizing effect of the secondary current I_2 by the amount necessary to produce the flux $\emptyset$ in the magnetic circuit. The curve $-I_2$ [curve (4), Diagram P] is then the component of the primary current required to neutralize the demagnetizing effect of the current in the rotor I_2. The primary current is then vector sum of I_E and $-I_2$ (see Diagram Q). If we assume that we are viewing the conditions in the motor from some point on the stator, then the curves of Diagram P and the vectors of Diagram Q will indicate the time variation, at primary frequency, as well as the space variation of the several variables.

In the following summary the significance of the vectors shown in Diagram Q have been tabulated:

E_1 is the e.m.f. impressed on stator.

R_1 is the stator resistance.

X_1 is the stator reactance.

I_E is the exciting current which is the vector sum of the magnetizing current Im and the core loss current I_{h+e}. I_E lags nearly 90 degrees behind the impressed e.m.f. E_1.

$\emptyset$ is the mutual flux, the vector $\emptyset$ is in the plane of the coil when it links the maximum flux.

E_2 is the e.m.f. induced in the rotor at standstill by the mutual flux $\emptyset$.

E'_2 is the actual e.m.f. induced in the rotor when there is a slip of S per cent. Then E_2 S equals E'_2.

E'_1 is the counter e.m.f. induced in the stator by the mutual flux $\emptyset$.

I_2 — the e.m.f. E_2 causes the current I_2 to flow. I_2 will have a time lag with respect to E_2 because of the rotor leakage reactance. — I_2 is the component of the stator current required to neutralize the demagnetizing effect of the rotor current I_2.

I_1 is the total stator current and is the vector sum of the current — I_2 and the exciting current I_E.

I_1R_1 is the stator resistance drop.

I_1X_1 is the stator reactance drop.

I_2R_2 is the rotor resistance drop.

I_2SX_2 is the rotor reactance drop at slip frequency.

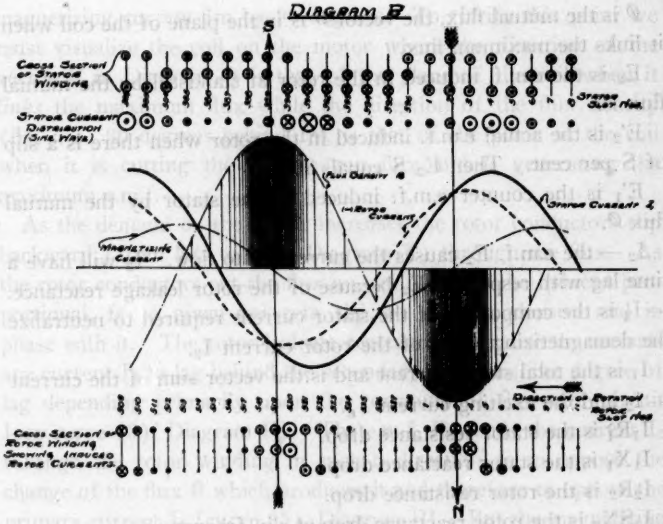
The impressed e.m.f. E_1 is equal to the vector sum of the counter e.m.f. E'_1 and the stator impedance drop I_1Z_1.

In Diagram R an attempt has been made to picture the approximate relative position of flux, poles, primary and secondary currents in the motor. The constants of the rotor and stator circuits were not available so the accurate locations of vectors could not be calculated. It will be noted on Diagram R that (a) the maximum value of the stator current leads the maximum value of the flux density by about 140 electrical degrees as on Diagram Q, (b) that the maximum value of the rotor current lags slightly behind the maximum value of flux density, (c) that the current in the rotor winding is in such a direction as to oppose the change of the flux which produces it and therefore is opposite to the stator current.

(*b*) 44-Pole connection.

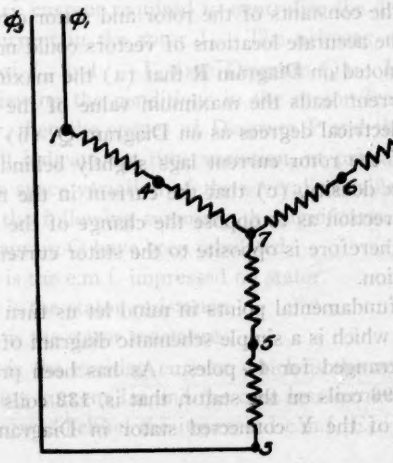
With the foregoing fundamental points in mind let us turn our attention to Diagram S which is a simple schematic diagram of the stator winding when arranged for 44 poles. As has been previously stated there are 396 coils on the stator, that is, 132 coils for each phase. Each leg of the Y connected stator in Diagram S

DIAGRAM R



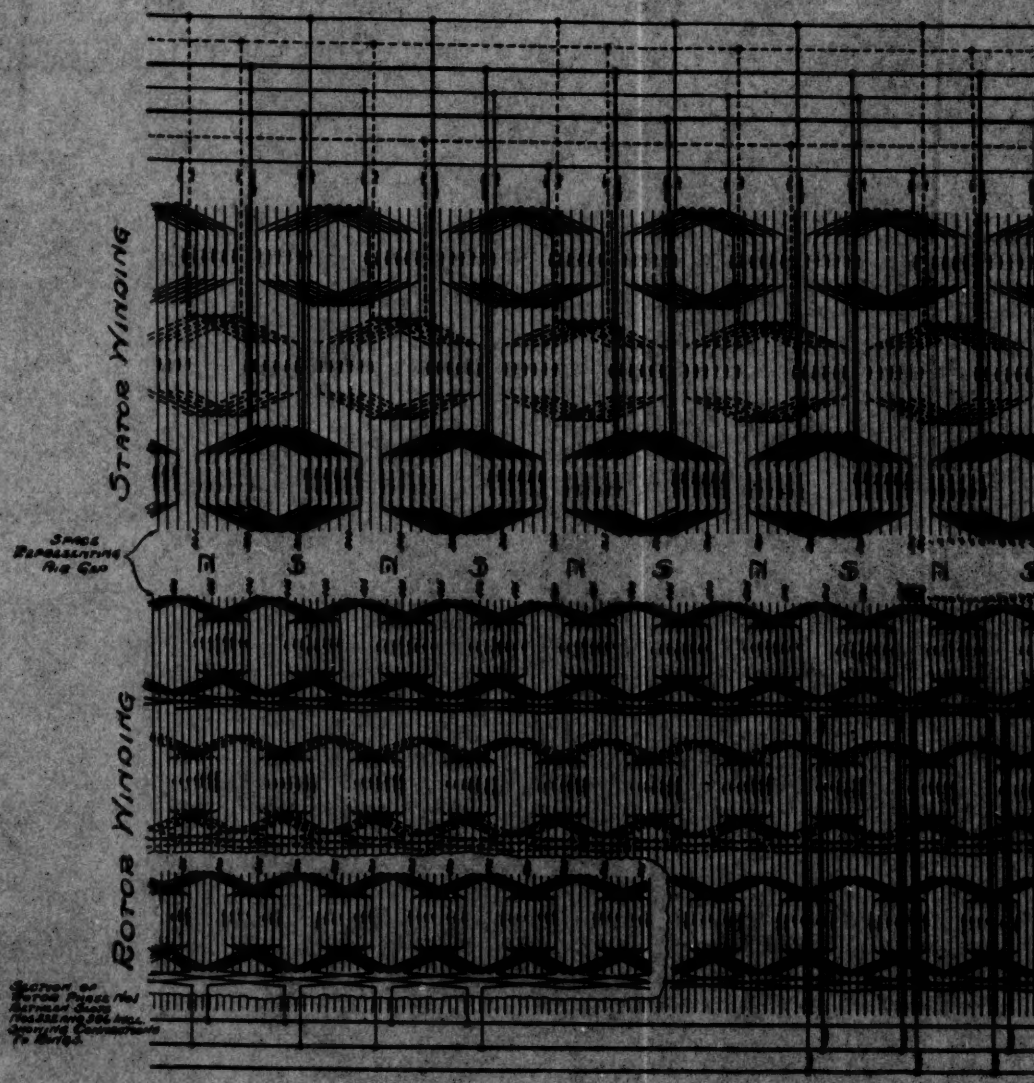
DIAGRAMMATIC REPRESENTATION OF THE VECTOR QUANTITIES IN THE WINDINGS OF THE LOADED INDUCTION MOTOR

DIAGRAM S



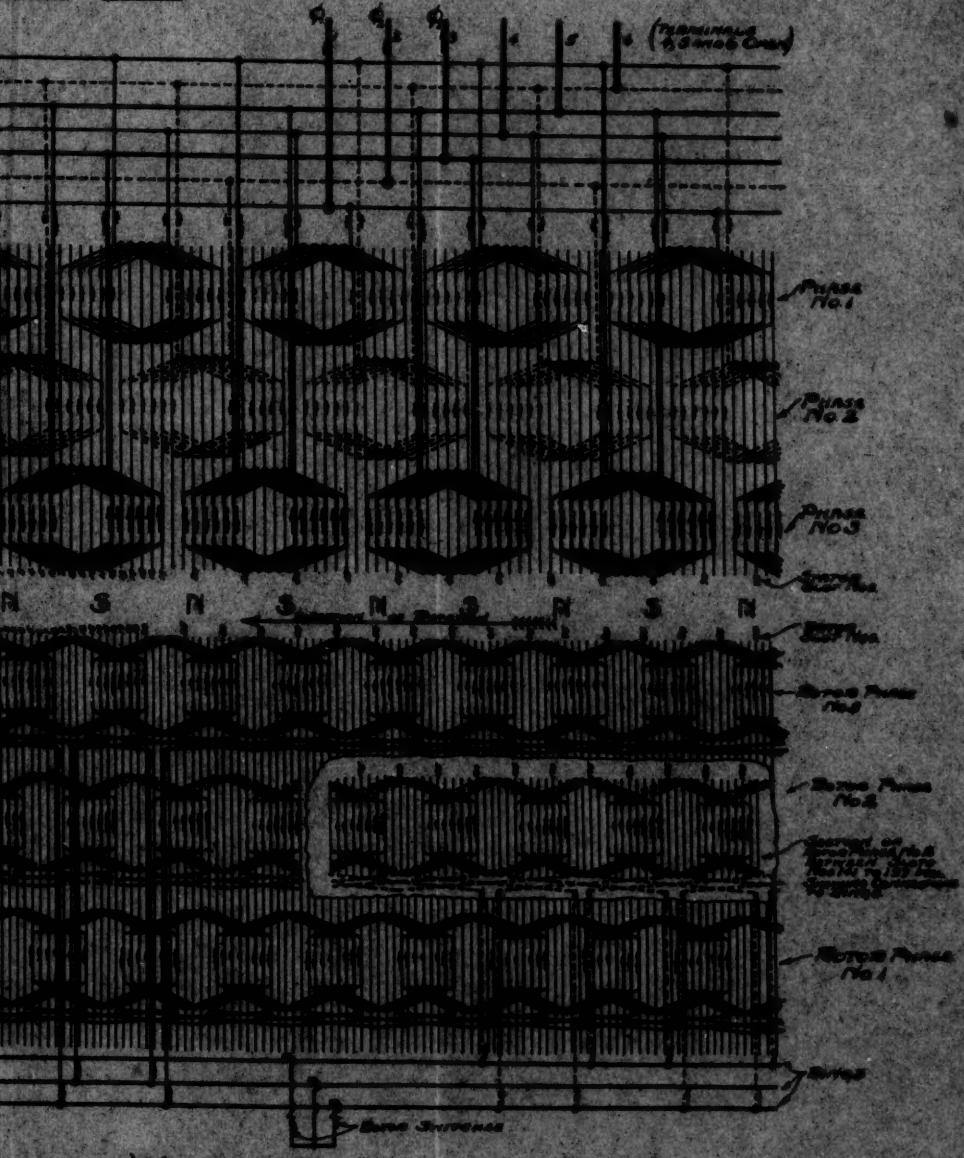
SCHEMATIC OF 44-POLE STATOR

DIAGRAM



MOTOR STATOR AND ROTOR
44-POLE CONNEC

BAM I



OTOR WINDINGS

NNECTION.

represents 132 coils connected in a certain way. In the 44-pole combination the current supply comes from the pole changing switches to the outside points, *i. e.*, 1, 2 and 3 of the legs of the Y connected stator. Now each section, as from point 1 to point 4, represents eleven groups of coils connected in parallel. Each group consists of 6 coils connected in series. There are then 6 times 11 or 66 coils in each section as from point 1 to point 4. Actually as will be seen later each point, *i. e.*, points 1 to 7 inclusive, is a ring to which the coil groups are connected. It will be noted that there are two sections to each leg of Y connected stator and that these sections are connected in series. Each leg of the Y connected stator on the 44-pole connection consists then of 2 sections of 11 groups each connected in series, the 11 groups of coils making up a section connected in parallel, each group made up of 6 coils connected in series.

Diagram T shows the wiring of the stator and the rotor of a main propelling motor together with the directions of the instantaneous stator currents, resultant poles, and the induced rotor currents for the 44-pole combination. A sufficient portion of the winding of both stator and rotor has been included to clearly show all the connections. In order to show the coil groups, their arrangement, and connection the phases have been separated. Actually, of course, the phases are wound one on top of the other, so that there are two conductors for each slot. The conductor in the bottom of the slot is shown as a broken line in Diagram T while the conductor on the top is shown as a full line. From the wiring diagram we can now see how six stator coils are connected in series to form a group and how 11 groups are connected in parallel between two rings, ring 1 to ring 4 for example, to form a section and how the groups of two sections, for example from ring 1 to ring 4 and from ring 4 to the neutral ring 7, are connected in series to form one leg of the Y connected stator.

The rotor winding, as will be noted from Diagram T, is a three-phase arrangement. In order that the high resistance squirrel cage winding (not shown in Diagram T) in the bottom of rotor slots will be effective during starting some method of opening the rotor phase winding must be provided. Accordingly, the so-called

rotor phases, are connected through slip rings across two rotor contactors which open the low resistance phase winding during starting when high torque is required. As will be noted from the wiring diagram, the rotor winding is a form of wave winding with four parallel paths around the rotor between any two slip rings. The wiring diagram shows how the connections are made from the rotor phase winding to the slip rings. The section of the rotor winding of phase 2 between slots 141 and 197 and the section of the rotor winding of phase 1 between slots 332 and 396 show that the three sets of connections to the slip rings are 120 degrees apart. It is interesting to note that each turn of the phase wound rotor coils consists of a solid copper bar insulated with four layers of one-half lapped mica tape and covered with a layer of linen tape with several treatments of varnish. The bars are formed at one end. The second end is formed after the bar has been pushed through the slot and is in the assembly position. The bars are then connected in groups in the proper sequence by soldered clips.

In the winding Diagram T the same instantaneous values for the stator currents was used as in the discussion of the fundamental principles, *i. e.*, the instant (X) on Diagram N where the instantaneous currents in phases 1 and 2 are one-half maximum positive and the instantaneous current in phase 3 is at its maximum negative value. It will be noted that the arrows showing the directions of the currents in the stator and rotor also indicate by their length the approximate relative values of the currents, *i. e.*, the current arrow in phase 1 of the stator is the same approximate length as a current arrow in phase 2, while the arrow in phase 3 is approximately twice the length of an arrow in either phase 1 or phase 2. Turning again to Diagram R which is a diagrammatic representation of the vector quantities in the windings of the loaded induction motor, we note the following:

1st, the currents in phases 1 and 2 are one-half maximum positive value while the current in phase 3 is at its maximum negative value.

2nd, that the flux density B is at its maximum value 90 degrees after the magnetizing current has reached its maximum value.

3rd, that the maximum value of the stator current leads the maximum value of the flux density by about 140 electrical degrees.

4th, that the maximum value of the rotor current lags slightly behind the maximum value of flux density and is in such a direction as to oppose the change of flux which produces it and therefore is opposite to the stator current.

5th, that the value of the stator current is at each instant the vector sum of the exciting current and the negative rotor current because the stator current must supply the exciting current as well as neutralize the demagnetizing effect of the rotor current.

6th, the induced rotor currents have the directions and are of an approximate relative magnitude as shown in cross section of the rotor winding in Diagram R.

It must be remembered that the angular displacements of the various vector quantities are merely approximately correct because the correct positions of the vectors could not be determined without knowledge of the rotor and stator constants. The approximate locations of the voltage vectors are shown on the vector Diagram Q.

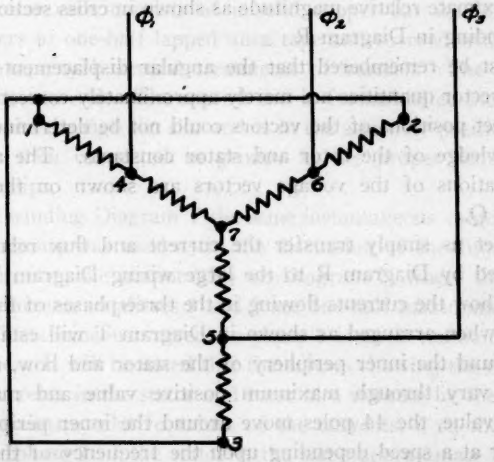
Now let us simply transfer the current and flux relations as determined by Diagram R to the large wiring Diagram T. We see now, how the currents flowing in the three phases of the stator winding when arranged as shown in Diagram T will establish 44 poles around the inner periphery of the stator and how, as these currents vary through maximum positive value and maximum negative value, the 44 poles move around the inner periphery of the stator at a speed depending upon the frequency of the stator currents. The flux from the 44 poles generates e.m.f.'s in the rotor phase winding which will cause currents to flow in the directions shown provided the motor rotor switches are closed. It will be noted from the directions of these currents in the phase winding that, unless the rotor switches are closed short circuiting the rings, no current will flow because the current paths are not closed circuits. With the rotor switches open the high resistance squirrel cage bars, which are adaptable to any number of stator poles, have currents induced in them which are more nearly in phase with the impressed e.m.f.'s and therefore produce the high torque required for starting. When the rotor switches close, as

the main field lever is moved from super to normal excitation, the rotor phase winding with lower copper loss, less heating, and greater efficiency, becomes effective.

(*c*) The 22-pole connections.

Diagram U is a simple schematic diagram of the motor stator when arranged for 22 poles. We notice that in this case the current supply from the pole changing switches comes to the center points of the legs of the Y, *i. e.*, 4, 5 and 6, and also that the out-

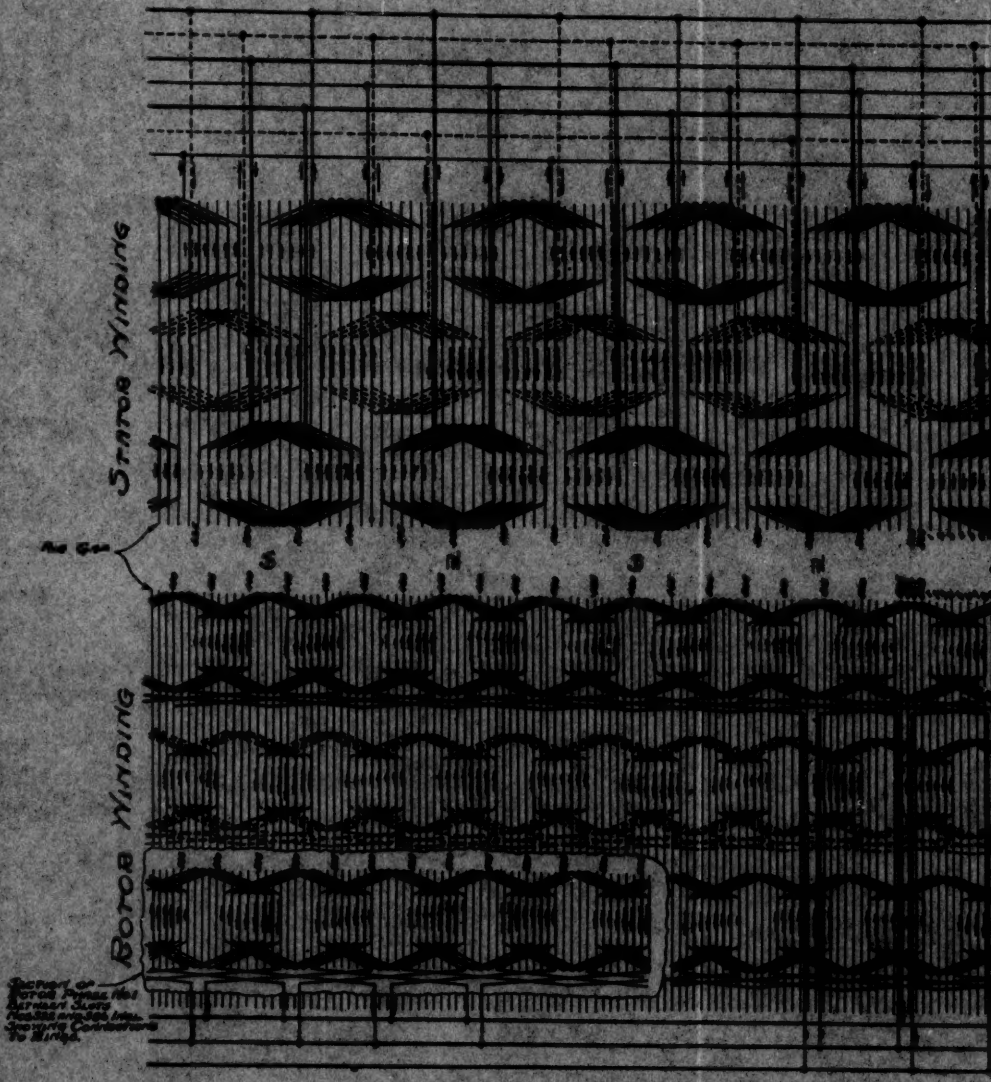
DIAGRAM U



SCHEMATIC OF 22 POLE STATOR

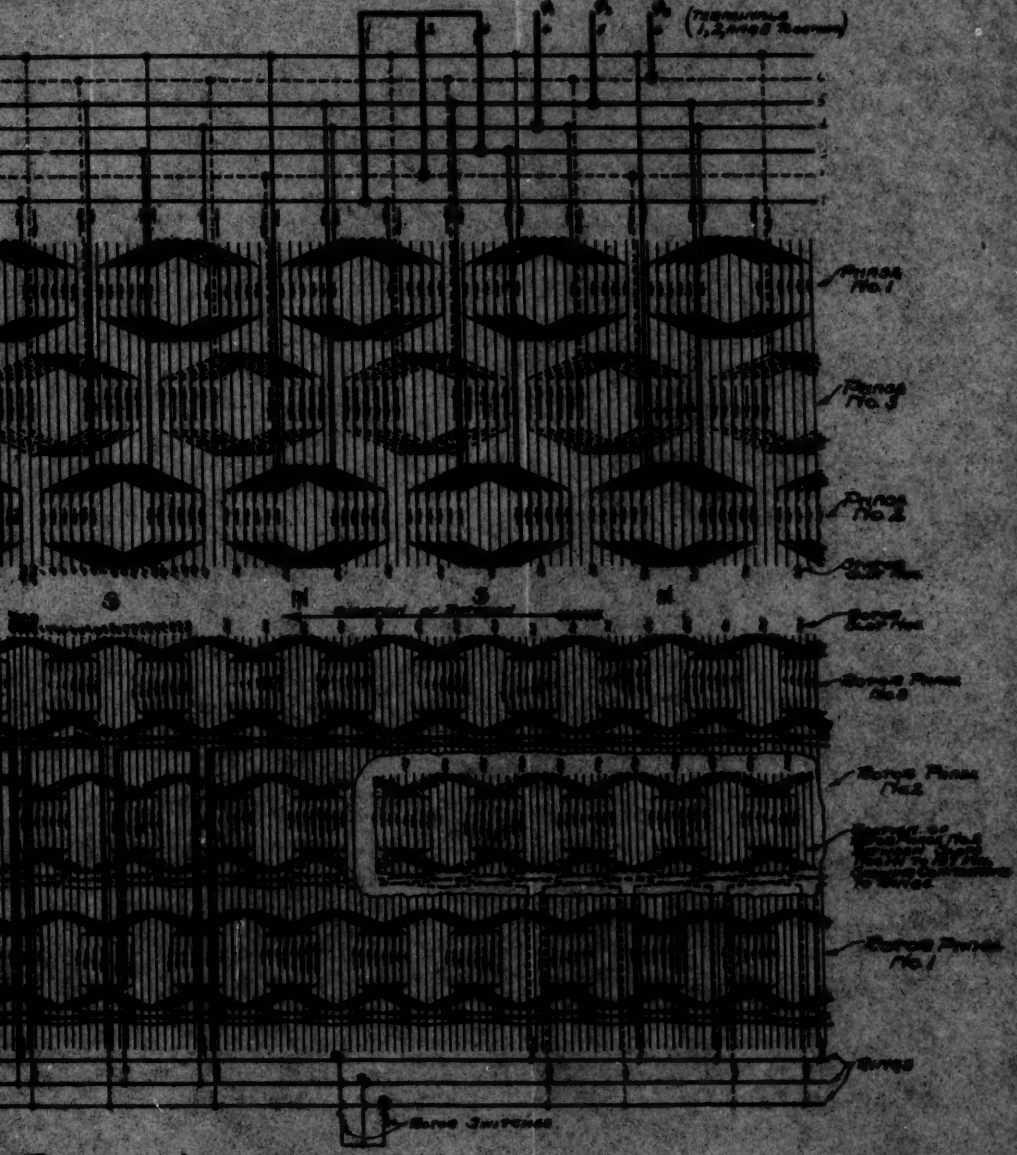
side points, *i. e.*, 1, 2 and 3 are connected together. As in the 44-pole connection, each section as from point 1 to point 4 in Diagram U represents eleven groups of coils connected in parallel and each group consists of 6 coils connected in series. There are then 6 times 11 or 66 coils in each section as from point 1 to point 4. Actually, as will be noted from Diagram V, each point, *i. e.*, 1 to 7 inclusive, in the simple schematic Diagram U is a ring to which the coil groups are connected. It will be further noted

Diagram



Motor Stator and Roto
22 - Pole Conn

GRAM V



ROTOR WINDINGS
CONNECTION

that each leg of the Y connected stator is made up of two sections as in the 44-pole connection, but that in the case of the 22-pole combination the sections are arranged in parallel instead of in series as for the 44-pole combination.

Diagram V shows the wiring of the stator and the rotor of a propelling motor together with the directions of the instantaneous stator currents, resultant poles, and the induced rotor currents for the 22-pole combination. As in the 44-pole wiring diagram a sufficient portion of the winding has been included to show all the connections. The phases have likewise been separated in order to clearly show the coil groups, their arrangement and connection. The conductor in the bottom of the slot is shown as a broken line while the top conductor is shown as a full line. The 22-pole diagram shows how the three-phase supply is connected to rings 4, 5 and 6 and how the outside points of the Y, *i. e.*, rings 1, 2 and 3 are connected together. We also see how six stator coils are connected in series to form a group and also how 11 groups are connected in parallel between two rings as for example from ring 1 to ring 4, Diagram V, to form a section. In the 22-pole combination the two sections of 11 groups each are connected in parallel to form one leg of the Y connected stator.

The arrangement of the rotor winding is exactly the same for 22 poles as for 44 poles. It will be noted, however, that the directions of the currents in the rotor phases are different in the 22-pole connection. As before, the lengths of the current arrows represent the approximate relative magnitudes of the induced rotor currents. Tracing the directions of the currents for example in rotor phase 3, we find that the voltages generated in two connected bars under adjacent poles cause currents to flow in opposite directions. For example the currents in bar 37 under a North pole flows in a direction opposite to the current in bar 58 under a South pole. However, the two bars are connected in series so that the currents flow in the same direction as far as their path through the rotor winding is concerned. Having traced the currents of one rotor phase through the four paths around the rotor we find that at the slip rings the currents from two paths are in one direction while the currents through the other two paths are in the

opposite direction. This then gives the currents complete paths around the rotor whether the slip rings are short circuited or not. In the 44-pole connection on the other hand we found that the currents per phase were in same direction in all four paths and therefore the rings had to be short circuited by the rotor switches in order that current could flow. It will be noted that the squirrel cage winding in the bottom of the rotor slots cannot give the high starting torque on the 22-pole connection because the form wound winding takes practically all of the induced current which is developed whether the slip rings are open or short circuited.

In the Diagram V the same instantaneous values of the stator currents were used as in the 44-pole diagram, *i. e.*, where the instantaneous currents in phases 1 and 2 are one-half maximum positive value and the instantaneous current in phase 3 is at its maximum negative value. The locations of the poles and the directions and approximate relative magnitude of the induced rotor current for the 22-pole connection are determined by the same method as used for the 44-pole connection.

C. PNEUMATIC CIRCUITS.

Although the pneumatic system is strictly not a part of the "propulsion wiring system of the *Lexington*" still the pneumatic circuits are so closely connected with the wiring circuits and successful operation depends to such a large extent upon the proper functioning of the various air engines and air valves, that a discussion of the propulsion circuits to be complete must necessarily include the pneumatic system. All the contactors mentioned in previous paragraphs, except the main field contactors, are operated pneumatically. To change the selection or "set up," to reverse, to shift from 44 to 22 poles or vice versa, to cut out a motor or a fan — all these operations are either entirely or partly accomplished by means of the pneumatic system.

Diagram W is a schematic arrangement of the various air engines, valves, and piping of the pneumatic system. An examination of the schematic diagram will show that the air supply is controlled through a system of valves and air cylinders to operate the following:

(1) The port and starboard generator contactors.
(2) The port and starboard reversing and fan motor contactors.
(3) The port and starboard pole changing contactors.
(4) The motor rotor switches in each motor room.

(*a*) Source of air supply.

The air supply for operating the pneumatic system may come from one of three sources: First, from the ship's pneumatic mains; second, from steam driven air compressors in the machinery spaces, and finally from two electrically driven railway type air compressors located in the main control room. The air pressure required for the system is about 90 pounds. The two railway type compressors are fitted with governors which are so arranged that they will start the compressors automatically when the pressure falls below 85 pounds and will stop the compressors when the pressure reaches 100 pounds. With three different sources of air entirely independent of one another, the possibilities of casualties due to a lack of air are very remote. In addition the reservoirs on the system hold enough air for about 12 operations of either set of air engines. As a last resort the hand operating gear is always ready for use.

(*b*) Reversing air engines.

The general constructional features of the reversing, pole changing, and generator air engines are practically the same. The air engines are operated by simply admitting air into one cylinder and exhausting air from the other cylinder. For example, to operate the reversing engines to reverse the ship, the hand levers on the main control board are moved to the "back" position which admits air into the "back" cylinders of the reversing engines through a system of valves and permits the air to be exhausted from the "ahead" cylinders.

The horizontal movement of the air engine piston rods is changed into the rotating motion necessary to move the contactor cam shafts by means of racks and pinions. The racks are secured to the piston rods and the pinions are secured to the ends of the various cam shafts. The rotating motion of the cam shafts causes the cams to open and close the various contactors as required.

The reverse levers on the main control board cannot be moved until the field control levers are in the "off" position. There is no time restriction between throwing off the field control levers and moving the reversing hand levers. It is, however, desirable to have a considerable time interval introduced between the throwing of the reversing levers and the actual opening of the reversing contactors in order that the current from the generators to the motors may die down to a certain predetermined value. On the electric drive battleships the under current relay is used to introduce this delay, but on the *Lexington* the speed of operation is controlled by means of a hydraulic timer.

(*c*) The hydraulic timer.

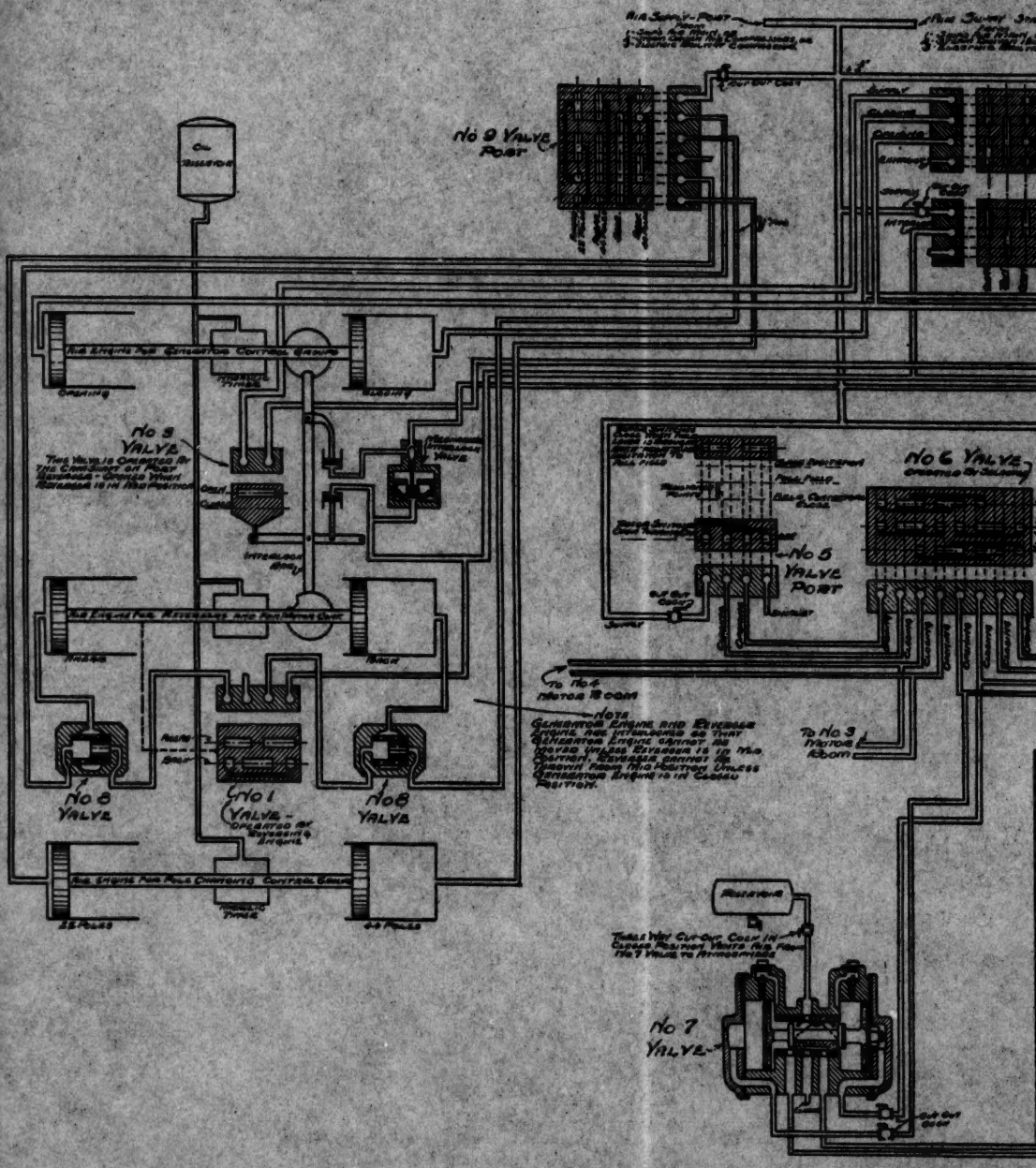
The levers on the main control panel can be rapidly set for reversal, but the hydraulic timer piston which is directly attached to the air engine piston as shown on Diagram W inherently introduces a delay of about 7 seconds before the contactors open. The time required to reverse the ship, *i. e.*, from the time the field lever is thrown to "off" position until the field is applied again to start the motors in the opposite direction, would be whatever time is required for manipulating the field levers, the reverse levers and the turbine speed and steam limit levers plus the 7 seconds delay introduced by the hydraulic timer.

Diagram X is a schematic diagram showing the cross section of a hydraulic timer and also a curve showing the speed characteristic of the piston of the air engine due to the action of the timer.

If we picture the piston moving from one side of the timer to the other we notice from Diagram X that the area around the piston varies due to the internal construction of the timer cylinders. When the area around the piston is greatest the liquid (80 per cent glycerin and 20 per cent water) permits the piston to travel at comparatively high speeds while when the clearance around the piston is small the speed is greatly reduced.

Let us assume for the time that air is admitted to the "ahead" cylinder of one of the reversing engines. From the position of zero speed, the piston travels very slowly (due to the throttling action of the hydraulic timer) until it reaches the position 2. At

DIAGRAM - W


No 9 Valve Post
No 3 Valve
No 6 Valve
No 5 Valve Post
No 4 Motor Room
No 3 Motor Room
No 8 Valve
No 1 Valve - Operated by Governing Engine
No 8 Valve
No 7 Valve

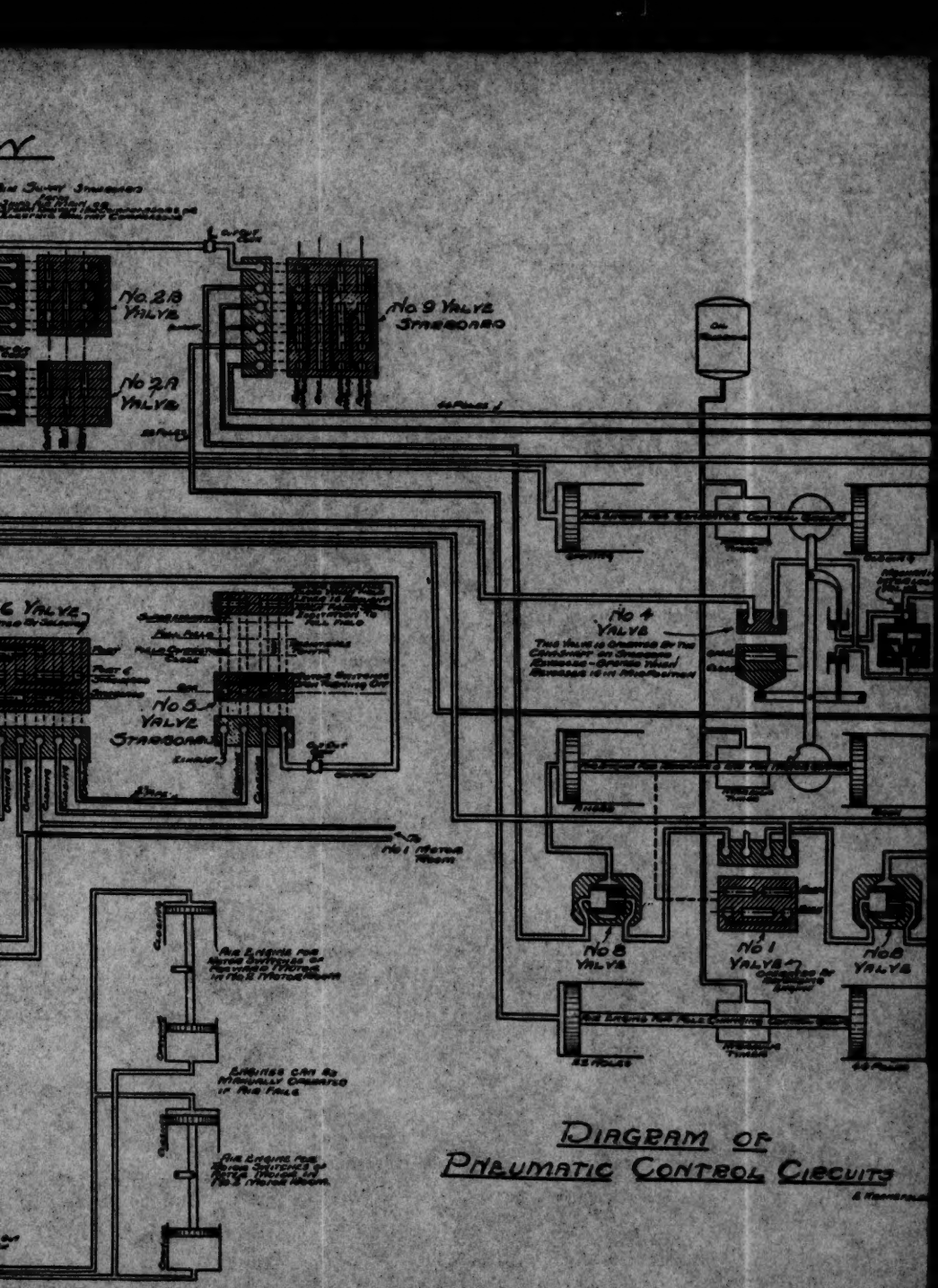
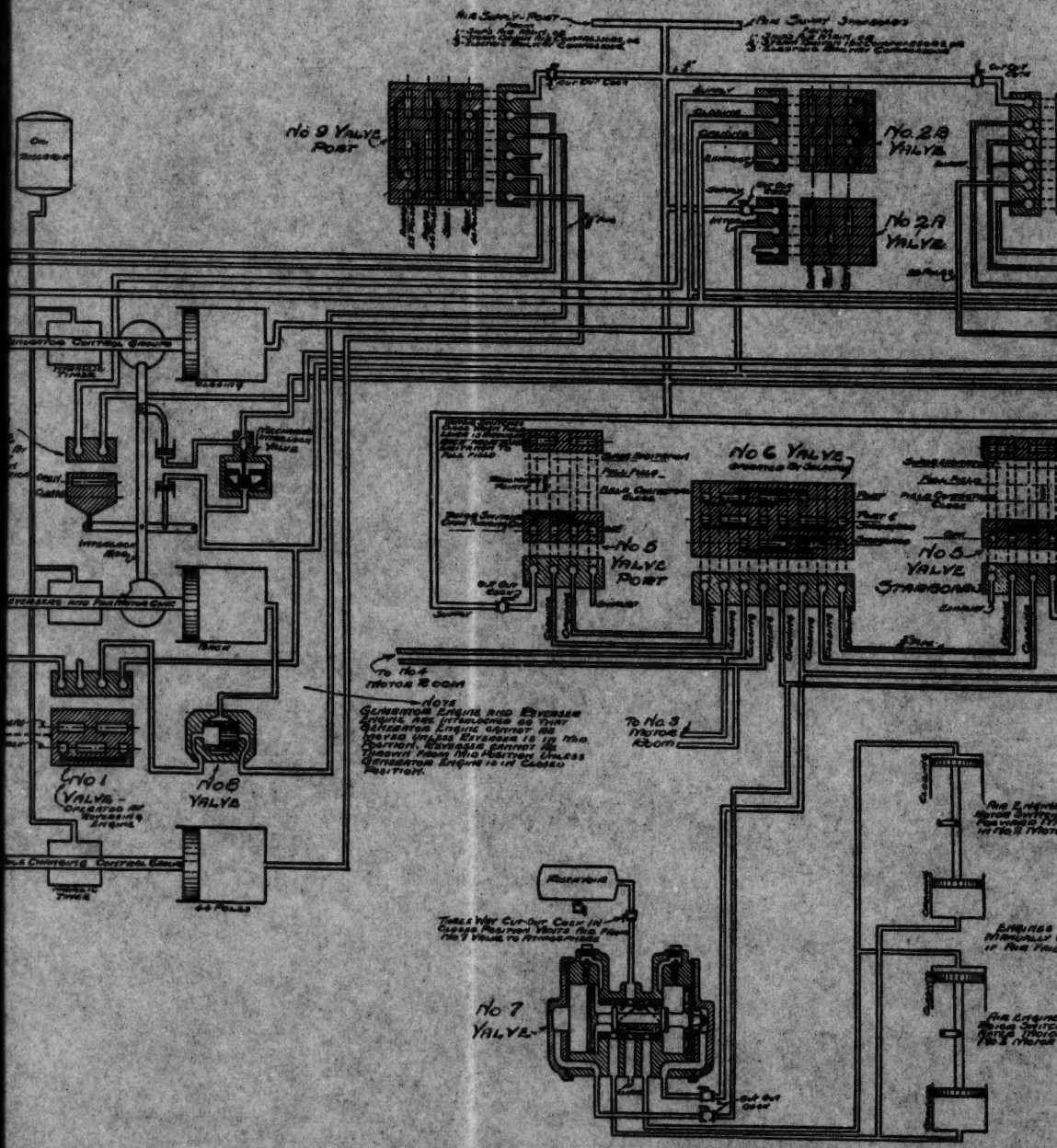

No. 2B Valve
No. 2A Valve
No. 9 Valve Starboard
No. 5 Valve Starboard
No. 4 Valve
No. 6 Valve
No. 1 Valve
No. 8 Valve
No. 1 Motor Room
Engines can be manually operated in Aux Fails
Diagram of Pneumatic Control Circuits

DIAGRAM - W


No 9 VALVE POST
No. 2B VALVE
No 2A VALVE
No 6 VALVE
No 5 VALVE POST
No 6 VALVE STARBOARD
To No 4 MOTOR ROOM
To No 3 MOTOR ROOM
No 1 VALVE - OPERATED BY REVERSING ENGINE
No 8 VALVE
No 7 VALVE
RESERVOIR
NOTE
GENERATOR ENGINE AND REVERSE ENGINE ARE INTERLOCKED SO THAT REFRIGERATOR ENGINE CANNOT BE MOVED UNLESS REVERSER IS IN THE CENTRE. REVERSER CANNOT BE THROWN FROM MID POSITION UNLESS GENERATOR ENGINE IS IN CLOSED POSITION.

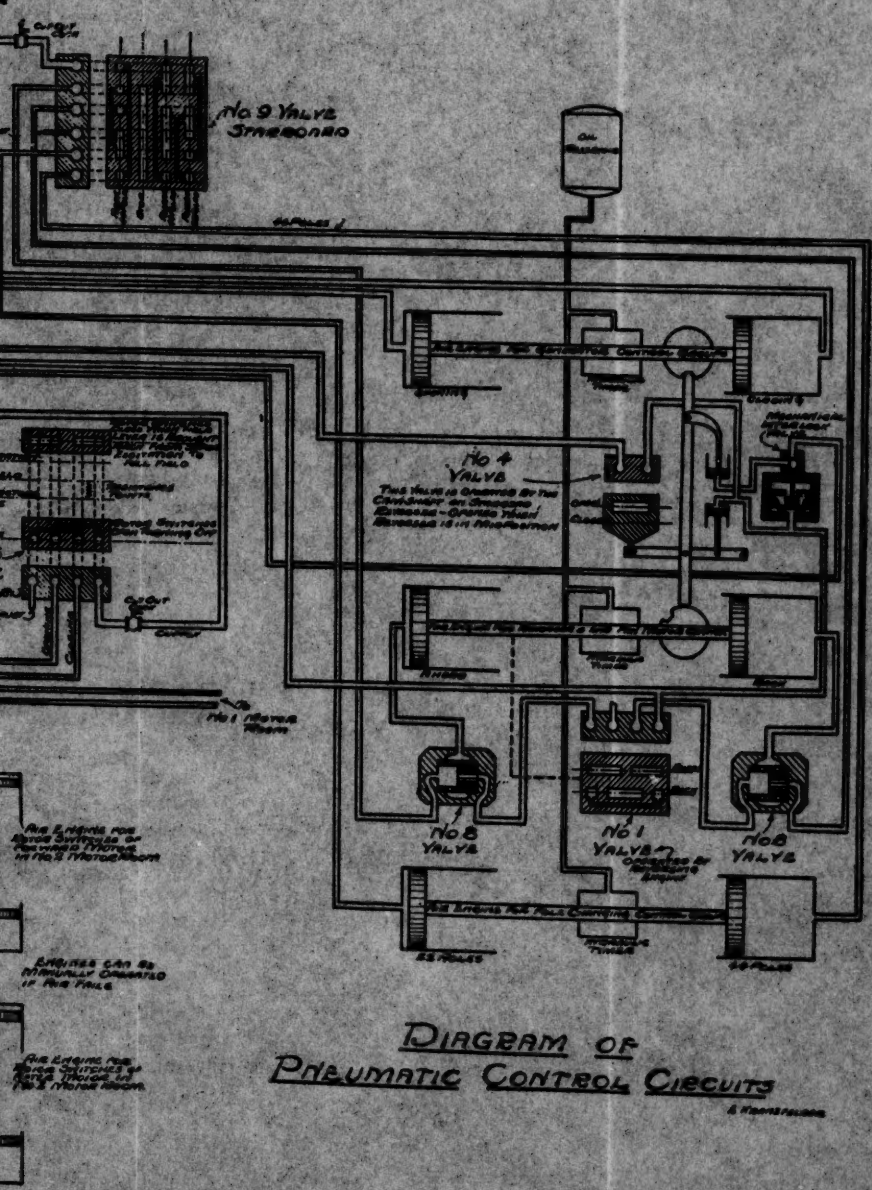
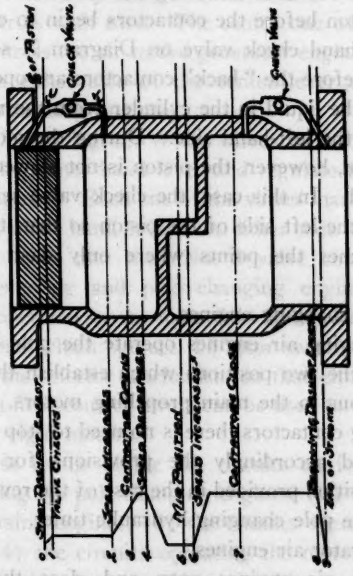


Diagram of Pneumatic Control Circuits

this position the "back" contactors start to open and while these contactors are opening the piston is traveling very rapidly in order that any arc that may be at the rupturing contacts and arcing horns of the reversing contactors (Diagram M) will be quickly extinguished. At position 4 the speed decreases and between position 4 and 6 there is a delayed action while the piston is travelling

DIAGRAM X



~HYDRAULIC TIMER~

through mid position where both the "ahead" and "back" contactors are open. This reduction in speed is necessary so that when it is desired to stop the reversing engine in mid position where both the "ahead" and "back" contactors are open, as when the generator switches are to be opened, the slowing down of the piston between positions 4 and 6 permits the mid position interlock bar shown in Diagram W to engage with the reversing engine piston rod locking the contactors in the open positions.

After the piston passes position 6 the speed increases again and at position 7 the "ahead" contactors close. As we approach the end of the stroke at position 8 the reduction in area around the piston slows it down and the hydraulic timer now serves as a dash pot to stop the piston slowly.

The check valves at both ends of the cylinder are connected with the by-pass pipes "U" and "V" to the openings into the cylinder wall. The purpose of the check valves is to assist in checking the speed of the piston before the contactors begin to open. For example, the left hand check valve on Diagram X serves to check the piston just before the "back" contactors are opened by checking the flow of the liquid in the cylinder from the right-hand side of the piston to the left-hand side. During the movement in the opposite direction, however, the piston is not slowed down at this part of its travel. In this case, the check valve serves to relieve the pressure on the left side of the piston so that it is not slowed up until it reaches the points where only dash pot action is required.

(*d*) The pole changing engines.

The pole changing air engines operate the pole changing contactors between the two positions which establish the 22-pole and 44-pole connections to the main propelling motors. In operating the pole changing contactors there is no need to stop the engines in mid position and accordingly the provisions for stopping the engine in mid position provided in the case of the reversing engines are omitted in the pole changing hydraulic timer.

(*e*) The generator air engines.

The generator air engines open and close those generator switches required for a certain selection or "set up." The generator air engines function similarly to the reversing air engines except that no provisions are made for stopping in mid position nor is the piston slowed down in the first part of its movement. In this case the hydraulic timer serves merely as a dash pot at the end of each throw.

(*f*) The air valves and circuits of the pneumatic system.

Our first impression from a glance at Diagram W is that the pneumatic circuits are involved and complicated, but if we examine the diagram a little more we find that the circuits are quite simple.

With three small hand levers directly in front of him and the two field levers, one on either side of the hand levers, the operator has practically complete control of all the switching required to change the selection, to change the number of poles, to reverse or go ahead, to excite the generator fields and to open and close the rotor switches. Under full power conditions, two good operators, one for each side, can control the above switching, while for speeds up to 19 knots when only one generator is in use one operator can handle the controls with ease. The center one of the three small hand levers controls the air to the generator engines while the two hand levers on either side of the center lever control the pole changing and reversing engines.

Turning again to Diagram W the valves marked 2A and 2B are controlled by the center hand lever and regulate the supply of air to generator engines. Each one of the valves marked 9 is operated by one of the small hand levers on either side of the generator engine control lever and each No. 9 valve controls the air to one set of reversing and pole changing engines. The valves marked 5 are operated by the main field levers and control the air to the air engines which operate the rotor switches. Valve No. 6 is attached to the selector mechanism so that either the starboard or port or both field control levers can be used to operate the rotor switches to all eight motors.

For clearness let us divide the circuits as follows: (1) The circuits operating the pole changing engines, (2) the circuits operating the reversing engines, (3) the circuits operating the generator engines, (4) the circuits operating the motor rotor switches. The interconnection between these circuits can be more readily explained when followed through in the above order.

(1) Each reversing and pole changing lever has four positions, *i. e.*, (*a*) stop, (*b*) ahead 44 poles, (*c*) ahead 22 poles, (*d*) back 44 poles. In Diagram W the stationary part of each one of the several valves is the part to which pipe lines are secured while the remaining part is the moving part attached to the operating lever. To shift from 44 poles ahead to 22 poles the two hand levers attached to valves No. 9 are placed in the 22-pole slots. The ports of valves No. 9 are then in line so that air from the supply line will enter the 22-pole cylinders and the air in the 44-pole

cylinders will be exhausted into the main control room. The piping in Diagram W clearly shows the air passage through the valves to the cylinders of the air engines which operate the 22-pole and 44-pole contactors.

(2) When the reversing and pole changing hand levers are placed from the "ahead" to the "back" slots and vice versa the operation of the reversing engines through the No. 9 valves is the same as for the pole changing engines except that the selective valve No. 8 is introduced into the circuit.

As has been stated in a previous paragraph the reversing engines must be placed in the mid positions whenever it is necessary to change the generator "set up" or selection and also whenever it is necessary to cut out or change the combination of fan motors or main propelling motors. With the reversing engines in mid position the main circuits are opened before any changes in the selections are made. To place the reversing engines in mid position the air is supplied to the cylinders from No. 2A valve instead of from the No. 9 valves. We have then two sources of air supply for the reversing engines and the purpose of the No. 8 valves is to provide an automatic cutoff for the air supply not in use.

In order then to place the reversing engines in mid position the No. 9 valves must be in the stop position and Nos. 2A and 2B valves must be in the open position. With these valves in the open position we can trace the air supply from 2A valve to the No. 8 valves through No. 1 valves and the pneumatically operated interlock shown in Diagram W. The moving parts of the No. 1 valves are connected to the piston rods of the reversing engines as shown by the dotted lines in the diagram. When a reversing engine moves, the corresponding No. 1 valve moves to such a position to provide a passage for the air supply to either the "ahead" or "back" cylinders of the reversing engines, depending upon which cylinder must be supplied so that the reversing engine will move to mid position. When the reversing engine is in the mid position the air supply to the top section of the interlock valve is cut off and air is admitted to the small cylinder which operates the interlock bar and forces it into the notch on the reversing engine piston rod. With the reversing engines interlocked in mid posi-

tion either the main motor or fan motor contactors can be cut out or the generator contactors can be opened and closed.

(3) In each case before the generator engines can move to the "open" position, the reversing engines, as previously described, must move to the "mid" position so that pneumatically operated interlock bars shown on Diagram W can be forced into the "mid" position slots of the reversing engines. At the same time that the interlock bars are forced into the mid position slots the pilot valves marked 3 in the port section and 4 in the starboard control section, are moved to their "open" position.

It will be noted in tracing out the pneumatic circuits to the generator engines that the valves 3 and 4 are connected in series. This connection assures that both starboard and port reversing engines are in the mid position and the interlock bars in place before any air can be admitted to the cylinders of the generator engines to open the generator switches. When both reversing engines are in the mid position and the generator switch hand lever is in the "open" position the path of the air to move the generator air engines to the open position is from the supply line through the ports of 2A valve, to valve No. 4, to valve No. 3, to the supply pipe line to 2B valve, and from 2B valve to the "opening" cylinders of the generator air engines. If the reversing engines are not in mid position the valves 3 and 4 remain closed and no air can get to the opening cylinders of the generator air engines. When the generator hand lever is moved to the "closed" position air is admitted to the closing cylinders of the generator air engines.

The generator engine hand operating lever and the reversing and pole changing hand operating levers are so interlocked mechanically that the operating lever for valves 2A and 2B must be in the neutral or "run" position before the pole changing and reversing hand levers which control the No. 9 valves can be moved from the "stop" position. And on the other hand, the No. 9 valve hand levers must be in "stop" position before the generator switch hand lever can move the valves 2A and 2B from the neutral position.

(4) The No. 5 valves on Diagram W are operated by the field control levers, one by the starboard lever and the other by the port

field control lever. With the 17 different combinations of generators supplying the eight main propelling motors there are some combinations which require the use of the starboard field control lever alone, as when C or D generator is being used. There are some set ups which require the use of only the port field control lever as when either A or B generator is being used and there are some set ups which require the use of both field control levers, as for example, when all four generators are used. It is evident that another valve is necessary which must be connected to the selector mechanism so that whenever there is a change in the "set up" which affects the field control lever a corresponding change must be made to the air connections to the No. 7 valves which regulate the air supply to the motor rotor switches.

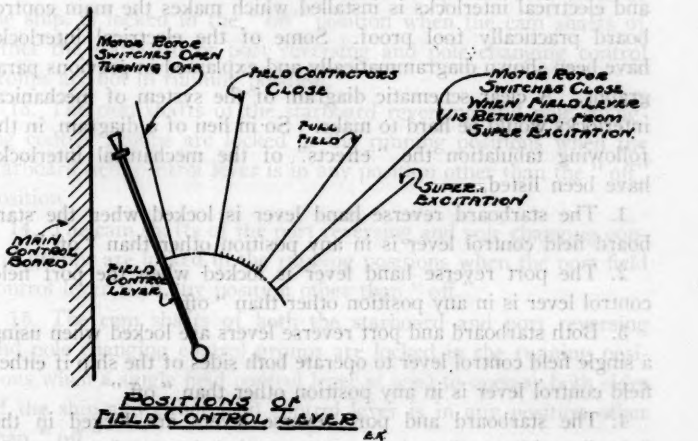
Valve 6 is provided to change the air supply from the No. 5 valves attached to the field levers whenever a change in selection is made. No. 6 valve has three positions, *i. e.*, one position controls the air supply to the rotor switches when the starboard field control lever is used, another position controls the air supply to motor rotor switches when the port field control lever is used, and the remaining position controls the air supply when both of the field control levers are in use.

When, for example, the port field control lever is moved to the "off" position, the No. 5 valve is moved to a position so that air from the supply line is admitted into the line marked "opening." The valve 6, which has been properly set by the selector for use with the port field control lever, admits air into the cylinders of the pilot valves 7 causing them to move and in turn admit air from the local air reservoirs in each motor room to the "opening" cylinders of the small air engines which operate the motor rotor switches. With the motor rotor switches open, the next time the field is applied the form wound phase winding is open circuited so that the high resistance squirrel cage bars giving high starting torque are effective.

When the motors are in step and the field lever is moved from "Super Excitation" back to "Full Field" the No. 5 valve moves to a position which moves the pilot valves 7 to a position to admit air to the "closing" cylinders of all the motor rotor switch engines

(see Diagram W_1, for positions of field control lever). The path of the air in this case is from the supply line through the pipe marked "closing" from the valve 5 on port side, to the selector valve 6. This valve is in the port side position and air is admitted to the "closing" pipe to the 4 motor rooms. In the motor rooms the air serves to throw the No. 7 pilot valves into the positions which will admit air from the reservoirs in each motor room into the "closing" cylinders of the motor rotor switch engines. When these switches close the form wound low resistance winding on the rotor is effective.

DIAGRAM W.


Motor Rotor Switches Open Turning Off
Field Contactors Close
Motor Rotor Switches Close When Field Lever is Returned from Super Excitation.
Full Field
Super. Excitation
Main Control Board
Field Control Lever

POSITIONS OF FIELD CONTROL LEVER

In Diagram W the pilot valve 7 and the motor rotor switch engines for one motor room are shown. The installations for all four motor rooms are identical. It will be noted that one pilot valve serves two motor rotor switch engines. Small ports in the cylinders of the pilot valves are exposed after each operation so that air can be admitted to the local reservoirs and keep them charged. The pilot valves are so constructed that they insure the completion of the movement whenever one of the motor rotor switches starts to throw.

(*g*) Mechanical interlocks.

The investigation of the general scheme of the control system impresses one with the relative simplicity of the entire system, when the large number of "set ups" and combinations of generators and motors and the flexibility of the control is considered. One is also impressed by the great ease with which one or two men can handle the controls and maneuver the vessel at high powers. The use of air to operate the six air engines, which in turn move a multiplicity of contactors, has aided greatly to make this simplicity of control possible.

In order to guard against any possible mistakes on the part of the operators in manipulating the controls a system of mechanical and electrical interlocks is installed which makes the main control board practically fool proof. Some of the electrical interlocks have been shown diagrammatically and explained in previous paragraphs. A clear schematic diagram of the system of mechanical interlocks would be hard to make. So in lieu of a diagram, in the following tabulation the "effects" of the mechanical interlocks have been listed.

1. The starboard reverse hand lever is locked when the starboard field control lever is in any position other than "off."

2. The port reverse hand lever is locked when the port field control lever is in any position other than "off."

3. Both starboard and port reverse levers are locked when using a single field control lever to operate both sides of the ship if either field control lever is in any position other than "off."

4. The starboard and port reverse levers are locked in the "stop" positions when the generator switch lever is in either the "open" or "close" position.

5. The generator switch lever is locked in the "run" position, except when both the starboard and port reverse levers are in the "stop" positions.

6. The starboard and port reverse levers and the generator switch lever are locked when the manual operating mechanism is engaged.

7. The starboard field control lever is locked in the "off" position when the starboard reverse lever is in the "stop" position.

8. The port field control lever is locked in the "off" position when the port reverse lever is in the "stop" position.

9. Either field control lever, if used to operate both sides of the ship, is locked in the "off" position when either reverse lever is in the "stop" position.

10. The starboard field control lever is locked in the "off" position when the cam shafts of the starboard reversing and pole changing control groups are not in running positions.

11. The port field control lever is locked in the "off" position when the cam shafts of the port reversing groups are not in running positions.

12. Either field control lever, if used to operate both sides of the ship, is locked in the "off" position when the cam shafts of either the starboard or port reversing and pole changing control groups are not in running position.

13. The cam shafts of the starboard reversing and pole changing control groups are locked in the running positions when the starboard field control lever is in any position other than the "off" position.

14. The cam shafts of the port reversing and pole changing control groups are locked in the running positions when the port field control lever is in any position other than "off."

15. The cam shafts of both the starboard and port reversing and pole changing control groups are locked in the running positions when a single field control lever is used to operate both sides of the ship and either field control lever is in any position other than "off."

16. The cam shafts of the generator switch control groups can only be moved when all cam shafts of the reversing control groups are in the mid positions.

17. The starboard main motor and fan motor cutout levers are locked when the cam shafts of the starboard reversing control groups are not in the mid positions.

18. The port main motor and fan motor cutout levers are locked when the cam shafts of the port reversing control groups are not in mid positions.

19. The levers for manually operating the control groups are normally disengaged. They are engaged when the selector lever for manual operation is moved from the "run" position.

20. The lever for manual operation can only be engaged when both reverse levers are in the "stop" positions and the generator switch lever is in the "run" position.

21. The selector wheel on the front of the control board, which is used to make any one of the 17 "set ups" or combinations, is locked when any cam shaft of the generator switch control groups is in the "close" position.

22. When one of the two generators of a machinery space is not in use, the selector mechanism cuts out the field contactors of the idle generator.

23. When the two generators of a machinery space are not in use, the selector mechanism does not cut out the field contactors for these generators.

Therefore, it is possible to test either of the two generators in an idle machinery space by means of the idle field control lever.

24. The generator switch lever cannot be moved from the "open" or "close" position while the generator switch cam shafts are moving between the "open" and "close" positions.

D. PROTECTIVE CIRCUITS, TESTING CIRCUITS, CONTROL CIRCUITS, WIRING FOR METERS, ETC.

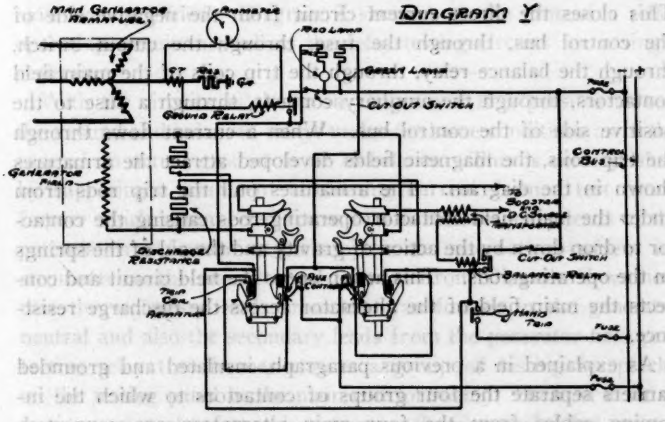
(*a*) Protective relay circuits.

In case of trouble, the simplest and quickest way to take power off all the main propulsion cables is to open the main field circuits of the driving alternators. Accordingly to afford protection against grounds or unbalanced circuits in the main generator and its connecting power circuits, the small trip coils of the main field contactors (see Diagram G) are energized which trip out the field contactors and open the field circuit.

In Diagram Y we have a simplified schematic diagram showing the protective relay circuits for tripping the field contactor of one main alternator. It must be remembered that there is a similar arrangement for each one of the four main alternators. In the diagram the main field contactors are shown in light lines. Nor-

mally, of course, these contactors are opened and closed manually by the operator when he moves one of the two main field control levers on the front of the main control board. However, as stated before, the field contactors can be tripped out by energizing their individual trip coils. These trip coils may be energized by either the ground relay or by the unbalanced relay shown in Diagram Y.

The ground relay is connected to the secondary side of a current transformer which is located in the grounded neutral of the alternator. An ammeter is connected in series with the secondary side of the transformer and will indicate at all times the amount of leakage current through the grounded neutral connection. There is one ammeter for each main alternator. These instruments are located on the main control board in the main control room. A resistance is placed in the neutral to ground connection of the main alternator which limits the amount of current which can flow through this circuit to ground. The purpose of the limiting resistance is to protect the windings and laminations of the generator from excessive injury in case of a heavy ground. The relay is set to operate at a current of 10 ampères or more through the neutral. When this amount of current flows the ground relay draws in the plunger shown in the schematic Diagram Y. This



PROTECTIVE RELAY CIRCUITS FOR TRIPPING FIELD CONTACTORS

closes a secondary circuit from the negative side of the control bus, through the cutout switch, through the trip coils of both field contactors (the coils being connected in parallel), through the auxiliary rods of the field contactors, through a fuse to the positive side of the control bus. The red lamp, in Diagram Y, is burning as long as the field circuit is closed but as soon as the contactors open the red lamp goes out and the circuit to the green lamp is completed through the auxiliary contacts on the operating levers. The green light indicates that the field circuit is open.

The balance relay shown in Diagram Y is connected to the secondary side of a transformer located in the generator field circuit. The transformer primary has a comparatively low resistance so that the excitation voltage will not be altered appreciably due to the drop through the primary of the transformer. Normally with only the direct current flowing in the field circuit no current is induced in the secondary of the transformer. However, when the three-phase power circuits are unbalanced more than a predetermined amount or in case of single phase operation, double frequency oscillations are superimposed upon the direct current flowing in the field circuit. These double frequency oscillations develop a current in the secondary of the transformer which attracts the plunger in the relay shown in the schematic Diagram Y. This closes the direct current circuit from the negative side of the control bus, through the fuse, through the cutout switch, through the balance relay, through the trip coils of the main field contactors, through the auxiliary contacts, through a fuse to the positive side of the control bus. When a current flows through the trip coils, the magnetic fields developed attract the armatures shown in the diagram. The armatures pull the trip rods from under the main field contactor operating rods causing the contactor to drop down by the action of gravity and the aid of the springs on the operating rods. This action opens the field circuit and connects the main field of the alternator across the discharge resistance.

As explained in a previous paragraph, insulated and grounded barriers separate the four groups of contactors to which the incoming cables from the four main alternators are connected. These barriers serve to localize and confine short circuits in case

they occur. With the neutral of grounded supports just above and below the bus bars an arc could not stretch an appreciable distance without going to ground, should the insulation of the busbar or contactor supports fail. If a positive ground occurs, the relay will function quickly opening the field circuit. The high tension cell of the main control room is well ventilated to carry off the gases that are developed in case of a ground or short circuit. A well grounded partition separates the high tension cell from the main control board where the operating personnel are stationed. Also the ventilating system of the control room is so designed that the space occupied by the operating personnel is under a slightly higher pressure than the high tension cell, so that any gases developed in case of a short circuit will go out through the exhaust trunk instead of leaking into the space occupied by the operating personnel.

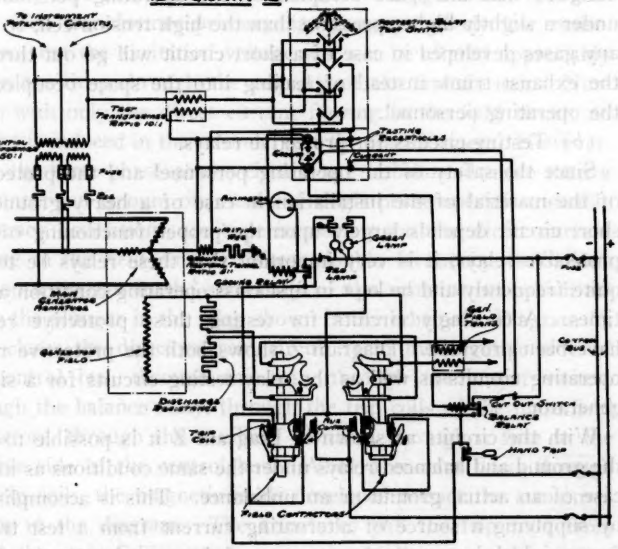
(*b*) Testing circuits for protective relays.

Since the safety of the operating personnel and the protection of the material of the installation in case of a heavy ground or short circuit depends largely upon the proper functioning of the protective relays, it is very important that these relays be tested quite frequently and be kept in first class operating condition at all times. Accordingly circuits for testing these protective relays have been provided. Diagram Z shows both the protective relay operating circuits as well as the relay testing circuits for a single generator.

With the circuits as shown in Diagram Z it is possible to test the ground and balanced relays under the same conditions as in the case of an actual ground or an unbalance. This is accomplished by supplying a source of alternating current from a test transformer which is connected across two of the secondary leads of the potential transformer which supplies the alternating current instruments on the main control board. From Diagram Z we note that the secondary leads from the transformer in the generator neutral and also the secondary leads from the generator field protective relay transformer run to a group of testing receptacles on the testing panel in the main control room. There is a set of testing receptacles for each generator, the receptacles for genera-

tors A and B are on the port testing panel, while those for generators B and D are on the starboard testing panel in the main control room. These testing receptacles serve as a junction point between the test transformer and the grounded neutral and generator field transformers. The receptacles also provide a means of connecting testing ammeters into the circuits at these points. The protective relay test switch has three positions, first, the vertical or "off" position, second, the left-hand position for testing the ground relay, and third, the right-hand position for testing the

DIAGRAM Z.



~ SIMPLIFIED CIRCUITS FOR TESTING PROTECTIVE RELAYS ~

E.K.

balance relay. There is a protective relay test switch for each main generator, located on the testing panels in the main control room.

It will be noted that when the test switch is in the "off" position the circuits between the test transformer and the protective relays

are open. This is the normal position of the test switch when the protective relays are not being tested. With the switch in the "off" position the circuits from the grounded neutral transformer to the ground relay and the circuits from the generator field transformer to the balance relay are in the normal operation condition and are exactly the same as the circuits shown in Diagram Y and explained under "protective relays." If now we wish to test the balance and ground relays, as before getting underway or before an additional generator is used in going to a two generator "set up," the field of the incoming generator is energized bringing the generator voltage up to about 1500 volts. If the switch lever is moved to the left or "ground" position, the circuit through primary of the test transformer will be completed. The circuit from the secondary of the test transformer to the ground relay will also be completed. The alternating current in the primary of the test transformer induces a current in the secondary which energizes the coil in the ground relay. This closes the D.C. circuit to the trip coils of the field contactors. The trip solenoids attract the armatures which trip the field contactors as explained in the paragraph describing the field contactors. An ammeter, which is mounted on the main control board, is permanently connected in series with the operating coil of the ground relay as shown in Diagram Z. This ammeter serves the double purpose of showing the magnitude of the leakage current in the neutral of a generator, as well as the magnitude of the A.C. supplied from the main alternating current cables when the ground relay is being tested.

When the operating lever of the test switch is moved to the right to the "balance" position, the operating coil of the balance relay is energized from the test transformer in the same way as described in the preceding paragraph for the ground relay test. If it is desirable to read the testing current when the balance relay is operated, a testing ammeter may be connected into the circuit by means of the receptacles on the testing panels.

(*c*) Phase failure relays in the fan motor circuits.

Diagram J showing the main propulsion wiring also shows the fan motor circuits. The bus connections for the fan motors are made between the generator switches and the reversing contactors. The fan motor contactors (one in each phase of a fan motor) are

operated by the same air engines which move the reversing contactors. The common cam shaft opens and closes the fan motor contactors simultaneously with the reversing contactors. The fan motor cutout mechanisms are similar to the main motor cutout mechanisms and each fan has an individual cutout lever on the main control board.

In order to protect the fan motors, an expulsion fuse is installed in each phase to the motors. A relay is also provided for each fan motor which indicates by means of signal lights on the main control board when the fan motor is operating normally and also when there is a failure or when the fan motor is operating single phase. Diagram AA shows the circuits to the phase failure relays and the indicating lights for the fan motors in one motor room. The diagram also shows the expulsion fuses in each phase to the fan motors. The two current transformers for each fan motor are connected as shown in the diagram. The secondaries of these transformers are connected to the operating coils of the phase failure relays and also to the meter testing receptacles. The relays have two sets of contacts. When, for example, normal current is flowing in each phase of a fan motor the bottom set of contacts on the relay for that motor is closed. This closes the D.C. circuit from the 240-volt control bus through the white indicating lamp showing the operator that the fan motor is operating normally. When there is a failure or when one of the expulsion fuses blows out while the fan motor is running and the fan continues to run single phase the relay will close the top set of contacts completing the D.C. circuit from the 240-volt control bus through the red indicating light showing the operator in the main control room that the fan motor is running single phase or that all fuses have blown out and the fan motor has stopped.

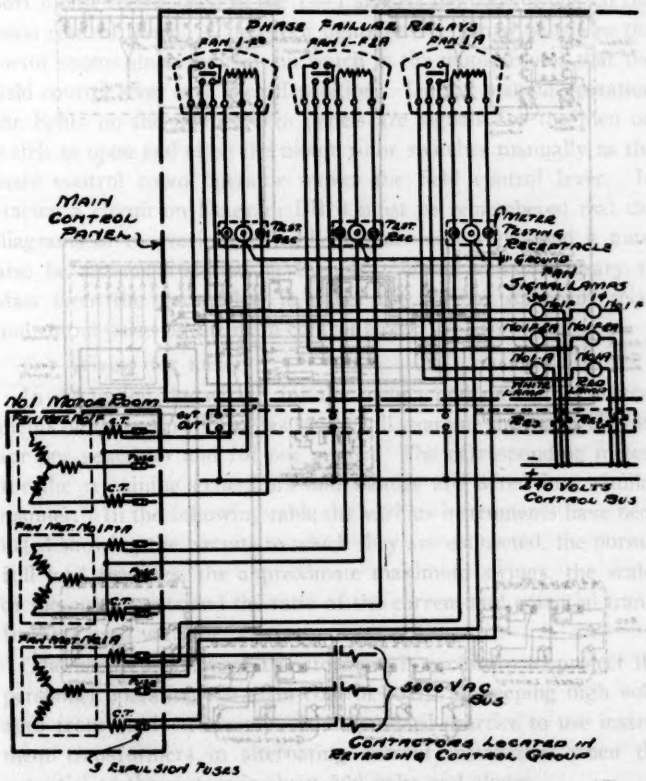
The fused cutout switches in the 240-volt D.C. circuits from the relays to the signal lights serve both as fuses and as cutout switches in these circuits.

The meter testing receptacles shown on Diagram AA are located on the testing panels in the main control room. These testing receptacles provide a means for reading the current in the fan motor circuits.

(*d*) Interlocking and signal circuits.

In some of the preceding paragraphs, the interlocking and signal circuits have been mentioned. Diagram BB is a simplified diagrammatic sketch of the signal and electric interlock circuits which connect the main control board with the machinery space switch boards, the motor room signal panels and the control room direct current switchboard.

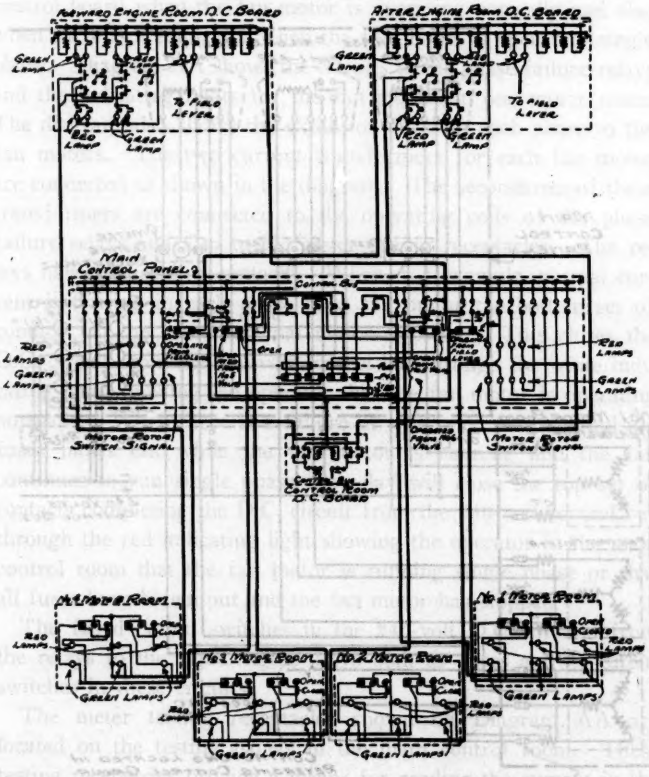
DIAGRAM AA



FAN MOTOR CIRCUITS

Diagram BB shows, for example, how the interlocks on the machinery space switchboards are energized when the respective field control levers are in the off position thus lifting the gravity interlocks so that the main excitation switches on the machinery space switchboards may be moved. Also how the various signal circuits are energized, for example, the circuits to the motor room

DIAGRAM BB



SIGNAL AND INTERLOCK CIRCUITS

2K

signal panels. When the field control levers are in the open positions as shown in the diagram the signal circuit for the port field control lever, for example, is as follows: From the (+) side of the control bus, through contact on valve No. 5, through the contact operated by the selective valve No. 6. In the position shown in the diagram No. 6 valve is set for one of the selections or "set ups" where both field control levers are used. From the contacts operated by valve No. 6 through the resistances shown the circuit leads to the green lights on the signal panels in the port motor rooms back to the (—) side of the control bus on the main control board. The green lights on the signal panels in the motor rooms show the men on watch in the motor rooms that the field control lever is in the off position. During manual operation the lights on the motor room panels are signals for the men on watch to open and close the motor rotor switches manually as the main control room operator moves the field control lever. In tracing a circuit on Diagram BB it must be remembered that the diagrams of the various switchboards are rear views and it must also be remembered that in tracing a circuit it is necessary to start from the control bus in either the port or starboard relay and cutout panel on the main control board.

(*e*) Wiring for meters.

In discussing the wiring for the meters and measuring instruments on the main control board we will examine the meter circuits for one generator and for one motor. The corresponding meters for the remaining generators and motors are wired in a similar manner. In the following table the various instruments have been listed showing the circuits to which they are connected, the normal full load readings, the approximate maximum swings, the scales of the instruments and the ratio of the current and potential transformers used with the instruments.

The use of instrument transformers is necessary to protect the personnel operating the main control board by keeping high voltages from the instruments. It is standard practice to use instrument transformers in alternating current installation when the potential of the system is about 500 volts and above.

Circuit	Normal Reading	Approx. Max. Swing	Instrument Scale	Remarks
Generator Volts	5000	6000	6500	Pot. Ratio 50:1
Generator Ampères	4620	11500	12000	C. T. Ratio 1200:1
Gen. Kilowatts	35200	44000	50000 Kw.	C. T. 1200:1 Pot. Rat. 50:1
Motor Ampères	2300	5750	3000	C. T. 600:1
Gen. Fld. Amps.	694	1200	1200	Used on 1200 Ampères 60 M. V. Shunt.
Generator Fld. Volts	172	280	350	.....
Excitation Indicator	Approx. 2	....	10–0	C. T. 1200:1 Pot. Rat. 50:1
Gen. Fld. Temp.	....	....	100°–580° F.	240 M.V. 750 Ampères Shunt–240 V.
Gen. and Mot. Temp.	....	....	70°–250° F. 20°–120° C.	Used on 240 V.
Turbine Speed	1770	2150	250	R. P. M. Operated by 2000 M. V. Magneto
Generator Speed	1770	....	2000	R.P.M. Frequency Indicator cal. in R.P.M.
Power Factor Ind.	.88	.90	.20–1.00	.....
Prot. Relay Coils	....	....	5 Ampères	Measures actual current in secondary of transformer in grounded neutral of Gen. C. T.5:1

The primary winding of the current instrument transformer consists of one or more turns and is connected in series with the main leads through which the current to be measured is flowing. The secondary winding is insulated from the primary turns and consists of a large number of turns, the number of turns depending upon the current to be measured. All the A.C. ammeters on the main control board, except the generator ammeters, have normal 5-ampère windings. The main generator ammeters have normal 10-ampère windings. The primary of the main generator current transformer consists of four ¼-inch x 4-inch soft drawn copper bars connected in multiple. These four bars connected

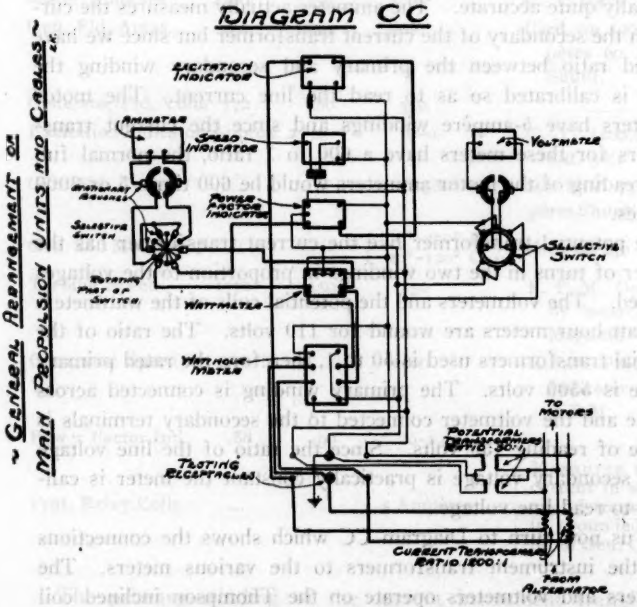
in multiple act as one turn for the primary winding. From the table we see that the ratio of the generator current is 1200 to 1. Since the generator ammeter used with the current transformer has a normal 10-ampère winding the secondary of the current transformer must have 1200 turns in order to give a full scale reading of 12,000 ampères. The ratio of a current transformer is usually quite accurate. The ammeter actually measures the current in the secondary of the current transformer but since we have a fixed ratio between the primary and secondary winding the meter is calibrated so as to read the line current. The motor ammeters have 5-ampère windings and since the current transformers for these meters have a 600 to 1 ratio, the normal full scale reading of the motor ammeters would be 600 times 5 or 3000 ampères.

The potential transformer like the current transformer has the number of turns in the two windings in proportion to the voltages involved. The voltmeters and the potential coils of the wattmeters and watt hour meters are wound for 110 volts. The ratio of the potential transformers used is 50 to 1, therefore the rated primary voltage is 5500 volts. The primary winding is connected across the line and the voltmeter connected to the secondary terminals is capable of reading 130 volts. Since the ratio of the line voltage to the secondary voltage is practically constant the meter is calibrated to read line voltage.

Let us now turn to Diagram CC which shows the connections from the instrument transformers to the various meters. The ammeters and voltmeters operate on the Thompson inclined coil principle. On board ship the use of an efficient damping method in the meters is especially important for in addition to the expected fluctuation of the load there will be added the roll of the ship and possibly to a less extent the effect of gunfire. Accordingly the main control board instruments are fitted with aluminum damping discs mounted on the instrument shafts and passing between the poles of two permanent magnets. The eddy currents set up in the damping discs effectively damp the oscillations of the moving systems of the instruments. Diagram CC shows the selective switch in the ammeter circuit by means of which it is possible to

read the current in any one of the three phases. In a similar manner it is possible to read the voltage across any phase by means of the selective switch for the voltmeter.

The *poly phase wattmeters* operate on the electro-dynamometer moving coil principle. The wattmeter really consists of two single



WIRING FOR INSTRUMENTS ON MAIN CONTROL BOARD ~ SHOWING METERS FOR ONE MAIN ALTERNATOR

phase elements acting on a single pointer. The single pointer reads the combined power of both phases. The wattmeters have 5-ampère current windings and 110-volt potential windings. The current coils and voltage coils are connected to the current instrument transformers as shown on Diagram CC. The scales are marked to read directly the total power in the circuit.

The *power factor indicator* also operates on the electro-dynamometer moving coil principle. Essentially the power factor indicator has a fixed current coil and a moving armature carrying two coils placed at right angles to each other. One coil is connected across the line and in series with a resistance while the other coil is connected in series with an inductance. The current in one coil will therefore be about 90 degrees out of phase with the current in the other coil. The ratio of the torques produced by the two coils of the armature gives a measure of the power factor of the circuit.

The *speed indicator* shown in Diagram CC is a frequency meter which has been calibrated in terms of alternator R.P.M. The meter is built on a resistance reactance principle. Diagram CC shows the frequency meter connected across two of the lines from the potential transformers. It will be noted that one lead enters a small box marked 3 on diagram. The current flowing through this lead divides in the resistance reactance box, one part passing through a non-inductive resistance to one of the frequency meter terminals, through one of the two coils in the meter and back to the lead connecting to the potential transformer. The other part of the current passes through an inductance (in the box) through the other coil in the meter and then unites with the first part of the current.

The principle of operation is that the strength of the current through the non-inductive resistance is independent of the frequency, while the current strength through the inductance varies inversely as the frequency. As the frequency increases less current flows through the reactive circuit causing a change in the interaction between the two coils of the meter. The ratio of the torques of the two coils, therefore, gives a true measure of the frequency of the circuit. Since both the reactive and resistive parts of the meter circuit are effected equally by a change in voltage, the meter is compensated for a variation in voltage.

The frequency meter described above has a disadvantage, in that there are no springs attached to the moving element, so when the voltage is removed the meter continues to read the same R.P.M. instead of reading zero. This is misleading at times.

Also the officer of the watch should know the turbine speed at all times, whether the field is energized or not and with this type of meter the reading means nothing when the field of the alternator is not energized. To indicate the turbine speed at all times, a magneto is geared to the turbine shaft which operates voltmeters, calibrated to turbine R.P.M., one located at the throttle of the respective turbine and one for each turbine located on the main control board. The magneto generator voltage varies directly as the speed of the turbine shaft, thus giving a reading of turbine speed at all times.

An excitation indicator for each generator is located on the main control board directly in front of the operator. The operator concentrates a good part of his attention on the excitation indicator, adjusting his generator field current to operate at a good power factor and still keep the motors in step. In a previous paragraph reference was made to the fact that for a given voltage and frequency a point is reached, as the torque of the load is increased, where the turning effort of the motors no longer increases and the motors "fall out of step," at the same time drawing a heavy current from the generator. It was also pointed out that the maximum torque which an induction motor can develop varies as the square of the impressed voltage. There is, however, another consideration and that is the power factor at which the motor is operating. From the standpoint of power factor alone it is advisable to operate at a high power factor because the higher the power factor, up to the maximum, the less the current demand. Since the amount of heat produced by the current in the motor and alternators' windings varies as the square of the current, the lower the power factor of the motor, the greater the current and the greater would be the amount of heat developed in the windings of the motors and alternators. A lower power factor then means lower efficiency. It is evident then, that in the operation of induction motors used for ship propulsion a compromise must be made between low impressed voltage giving high power factor and low pull out point on one hand and high impressed voltage giving high pull out point and low power factor on the other hand.

In the excitation indicator the operator has an instrument which will show him when he is operating at the best power factor and

still keep his motors in step. Essentially it consists of a current element and a voltage element each acting on a single shaft to which the indicating pointer is attached. The actual moving parts attached to the indicating shaft are two soft iron vanes placed at such an angle that the torques exerted by each are in opposition. One soft iron vane is under the influence of the current element while the other vane is under the influence of the voltage element. Each vane tends to move into such a position as to conduct the maximum number of lines of force. The stationary coil of one of the elements is connected to carry a current proportional to that in the main motor cables while the coil in the other element is connected across one phase to give a voltage proportional to that in the main lines. An impedance coil having a large number of turns is inserted in series with the voltage element. The resistance of the impedance coil being negligible, the reactance effect of the coil limits the current flow through the voltage element. For any given voltage the current in the voltage element containing the impedance coil is determined by the speed of the alternator. With increasing speed and constant field excitation, the voltage generated is increased in proportion to the speed and the frequency of the circuit also increases in proportion to the field. With constant field excitation the current flowing through the voltage coil of the excitation indicator will not be increased by an increase in voltage produced by an increase in alternator speed because the increase in voltage will be offset by the increase in the reactance of the impedance.

If now we consider the condition where the speed is kept constant and the field excitation is increased, the resultant increase in voltage will increase the current through the voltage coil because in this case the frequency remains unchanged and there is no compensating effect on the part of the reactance coil. The current in the voltage element is therefore proportional to the generator field excitation. On the other hand the current through the current element is proportional to the load on the motor. The vane of the current element tends to move the pointer of the excitation indicator to the right when the load on the motor increases while the vane under the influence of the voltage element tends to move

the pointer to the left when the field excitation of the alternator is increased. A balanced condition is obtained at some point on the scale. An arbitrary scale on the instrument is used on making comparisons for various operating conditions. When the pointer moves into the red section at the right of the scale it indicates that the motors are about to " fall out of step."

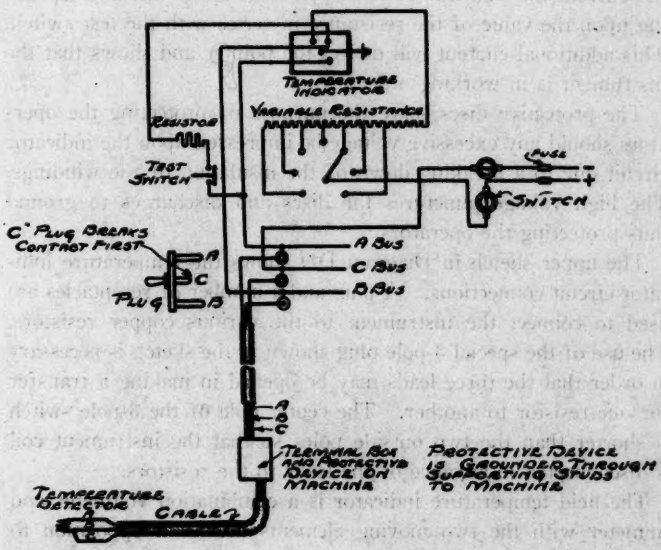
(*f*) Temperature indicators.

It is very important that the officer of the watch should know the temperatures of the alternator and motor windings at all times, in order that the safe operating limits are not exceeded and also so that the ventilation and circulating water through the air coolers can be adjusted for efficient operation. The temperature indicating system installed on the *Lexington* is similar to the system used quite extensively in commercial practice. A number of copper resistors were placed in some of the stator slots of the motors and alternators when they were built. These resistors are in close contact with the insulation in the warmest parts of the machines. The copper resistors reach the same temperatures as the adjoining windings and the temperature of the resistors is determined by the variation of their resistances.

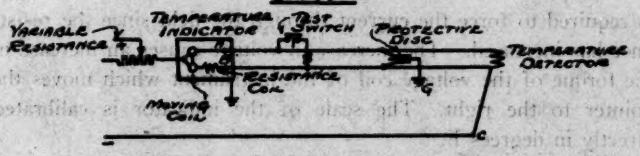
The lower sketch of Diagram DD is a schematic of the circuit to one resistor. The temperature indicator is a differential D'Arsonval galvanometer, that is, the two moving coils are connected differentially, one coil causing the needle to move to the right and the other coil tends to move the needle to the left. The positive supply is connected to the meter through a variable resistance which can be adjusted to compensate for a change in the voltage of the supply circuit. The current divides in the instrument, one part flowing through one of the movable coils and out along the upper lead from A, through the copper resistor in the alternator or motor stator and back to the negative of the operating bus through the lead from C. The other part of the current passes through the other movable coil marked "R," along lead B to the temperature detector or resistor and back along lead C.

The current in lead A depends upon the resistance of temperature detector in the machine, while the current in lead B remains the same. The difference in the torques of the two coils, due to the

DIAGRAM DD



SCHEMATIC OF TEMPERATURE INDICATING CIRCUIT



TEMPERATURE INDICATING CIRCUITS

difference in the currents in the coils, cause the pointer of the instrument to move along a scale which is graduated in degrees F.

Closing the self-opening test switch shown in the diagram causes the current through lead A to be increased by an amount depending upon the value of the resistance in series with the test switch. This additional current will deflect the pointer and shows that the instrument is in working order.

The protective discs serve the purpose of protecting the operators should any excessive voltage be impressed upon the indicator circuit due to a breaking down of the insulation of the windings. The high voltage punctures the discs and discharges to ground thus protecting the operators.

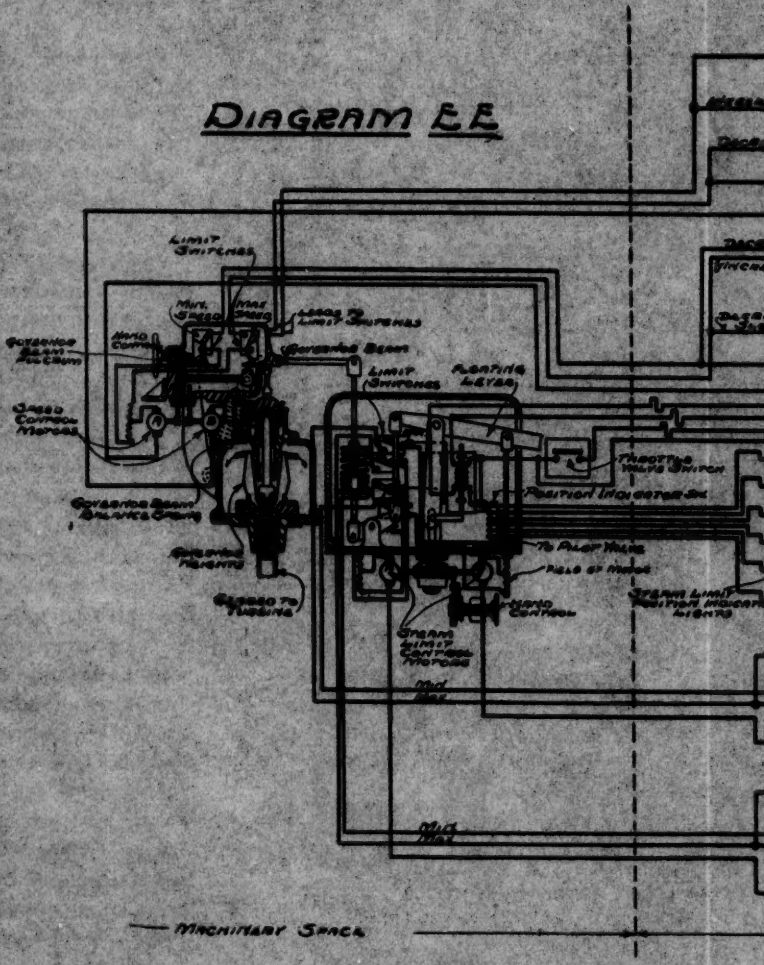
The upper sketch in Diagram DD shows the temperature indicator circuit connections. A plug and a number of receptacles are used to connect the instrument to the various copper resistors. The use of the special 3-pole plug shown in the sketch is necessary in order that the three leads may be opened in making a transfer for one resistor to another. The center pole of the 3-pole switch is shorter than the two outside poles so that the instrument coil is disconnected in advance of the leads to the resistors.

The field temperature indicator is a combination voltmeter and ammeter with the two moving elements acting in opposition to each other. The current element of the indicator is connected across a shunt which is in series with the alternator field while the voltage element is connected across the field winding. As the field temperature increases for a certain field current, more voltage is required to force the current through the field since the resistance has increased. The increase in voltage causes an increase in the torque of the voltage coil of the instrument which moves the pointer to the right. The scale of the indicator is calibrated directly in degrees F.

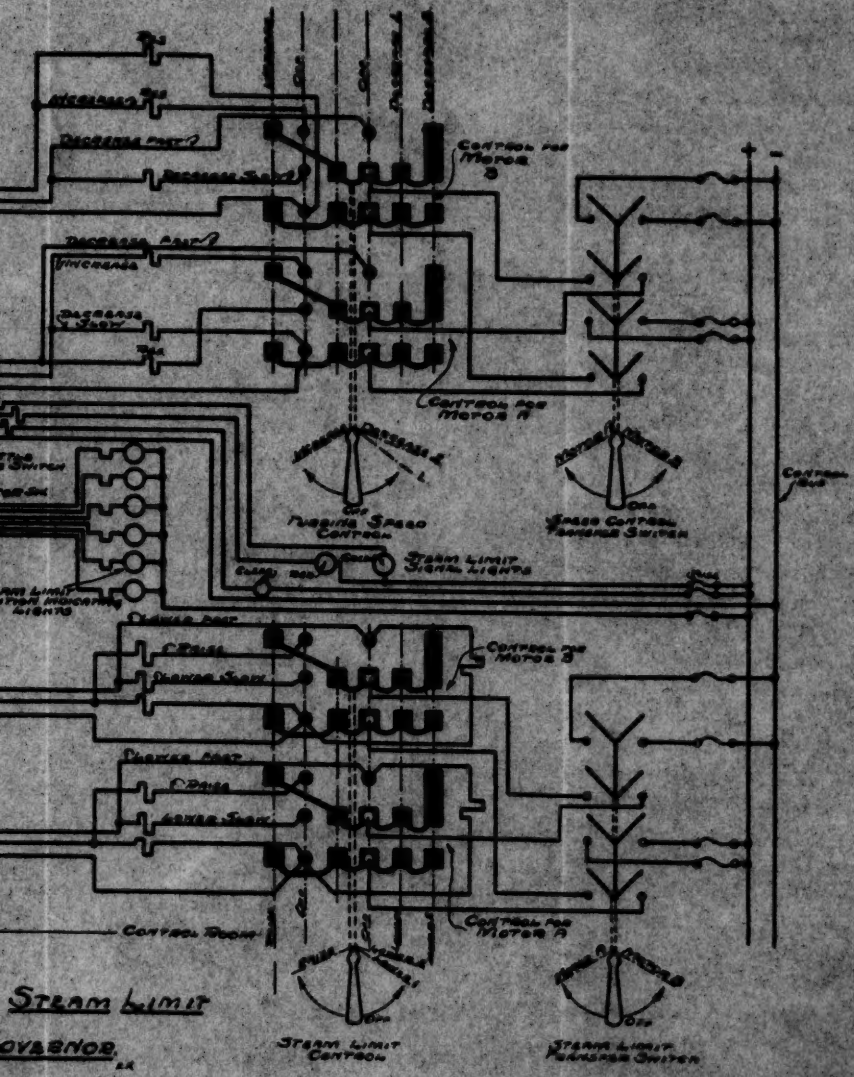
(*g*) Speed and steam limit control circuits.

In a previous paragraph reference was made to the fact that it was characteristic of an induction motor to operate within a few per cent of a certain given speed with a load varying through wide limits. This characteristic of the induction motor, when the motor is used for ship propulsion, presents a problem in the control of the steam to the turbo alternators. Under certain conditions, as

DIAGRAM E.E.



WIRING DIAGRAM FOR SPEED AND STEAM CONTROL FOR ONE TURBINE GOVERNOR


Control for Motor B
Control for Motor A
Control Bus
Steam Limit Signal Lights
Control for Motor B
Control for Motor A
Control Room
Steam Limit
Steam Limit Control
Steam Limit Reverser Switch

when the ship is in a heavy sea or when making a turn, the speed of the propellers remains practically constant while the load on the generators varies through wide limits. The governor of the turbine in trying to meet the large increase in the demand for power from the alternator opens additional control valves and draws heavily upon the steam supply. For example when the ship is making a turn, let us say to port, the load on the starboard motors decreases while the load on the inboard or port motors is greatly increased. On a ship driven by turbines or reciprocating engines in making a turn, the inboard or port engines in a turn to port with no change in the throttle setting, slows down while the outboard or starboard engine speeds up. On the electric driven ship, however, the induction motors tend to run at nearly constant speed and consequently the demand on the turbo generator supplying the inboard shafts is greatly increased.

To limit the amount of steam which a turbine can take in trying to supply a large demand, as the demand of the inboard shafts in making a turn, a steam limit stop on the governors is provided. This steam limit stop is adjustable and the setting is controlled by the operator in the main control room. When the governor, in trying to supply a large demand for steam, comes up against the steam limit stop, no additional steam can be admitted to the turbine. The steam limit stop is so constructed that it does not limit the governor from reducing the supply of steam to the turbine.

Let us now examine Diagram EE which shows the wiring circuits for the turbine speed control and steam limit control. Looking at the turbine speed control circuits first, we find there are two small motors which are geared to the lead screw on the governor. Rotation of the motor armatures changes the position of the sliding nut on the lead screw which in turn changes the tension of the governor spring thus changing the governor setting and consequently the turbine speed. The speed setting of the governor is then increased or decreased, depending upon the direction of rotation of the motors. Although both motors shown in Diagram EE are mechanically connected to the lead screw of the governor mechanism only one motor is electrically connected at one time,

the other merely serving as a "standby" in case of a casualty to the first motor. Each motor is equipped with two fields to provide for operation in either direction.

Let us trace the circuit to one of the speed control motors beginning at the control bus on the right-hand side of Diagram EE. We first meet the speed control transfer switch which is simply 4-pole double throw switch used to energize either one of the two speed control motors depending upon the position of the lever. This switch is mounted directly behind the main control board while the lever is in front of the board within easy reach of the operator. The speed control switch is used by the operator to effect gradual adjustments in the setting of the operating governor of the turbine. In looking at the wiring Diagram EE of the turbine speed control switch it must be remembered that the switch is of circular construction and that the heavy black contacts move with the switch handle while the small round dots represent the stationary contacts. The switch is turned through a small arc in one direction to increase the governor setting and in the other direction to decrease the setting. A spring is attached to the shaft of the switch which always returns the switch to the "off" position except when the switch is placed on the full decrease position. The switch remains in the full decrease position unless moved manually to the "off" position. The governor setting is decreased as quickly as possible when the lever is in the full decrease position. Resistors in series with each half of the field of each pilot motor serve to reduce the motor speeds so that fine adjustments of the governor settings can be made. It will be noted that one resistor is connected across the motor armature. When the switch is moved to the "off" position the motor continues to rotate. The brake resistor tends to load the motor which is now acting as a generator and brings it to rest in a few revolutions. When increasing speed the action of the governor spring is sufficient to bring the motor to rest quickly.

Mounted on a small bakelite panel in the governor housing are four limit switches, two for each pilot motor. These switches serve to stop the pilot motor in use, when the nut on the lead screw reaches either end of the travel.

With but a few exceptions the circuits for operating the steam limit motors are same as those for the turbine speed control motors. The steam limit switch does not stay in the "full" decrease "position," but springs back to the "off" position each time. There are braking resistors connected in both the increase and decrease motor circuits so that the motor will come to rest quickly when operating in either direction.

The position indicator and signal lamp switch is an additional feature of the steam limit system. There are two sets of contacts which are closed and opened by the movement of the governor. When the governor setting is too far away from the steam limit setting the lower set of contacts are closed through a copper disc and the green light on the main control board lights up. On the other hand when the governor approaches the setting of the steam limit the upper set of contacts are closed and a red light on the main control board shows the operator that he is approaching the steam limit setting. When within the proper range for normal steaming, the spring between the two copper discs closes both sets of contacts and both the red and green lights will light up.

In addition to the above indicating system there is another system on each turbine steam limit mechanism which shows the operator in the main control room the position of his steam limit at all times. This system consists of six electrical stationary buttons and a brush or contact which moves up and down with the steam limit stop. The brush completes the circuits to the signal lights on the main control board. The brush is so constructed that in passing from one contact button to the next it is possible to bridge two contacts at a time. Two of the signal lights will light under this condition. With this arrangement the six lights will show eleven definite points of travel of the steam limit stop, *i. e.*, six when but one light is lighted at a time and five positions in which two lights are lighted at a time. This system of lights serves to indicate the position of the steam limit stop at all times.

The operator usually keeps his steam limit stop in such a position that both the red and the green lights on the main control board are lighted. However, in cases demanding a large amount of power, as in reversing, the steam limit setting must be raised so

that additional steam can be admitted into the turbine. The operator must be careful not to raise the limit stop too quickly for there is the danger of carrying over water from the boilers with sudden large demands for steam.

Before closing the writer wishes to express his appreciation for the assistance given by Lieutenant H. L. Dodson in proof reading this article.

A STUDY OF THE POSSIBILITIES OF THE HIGH SPEED, TWO-CYCLE, DOUBLE ACTING ENGINE; WITH PARTICULAR REFERENCE TO ITS AVAILABILITY FOR HIGH POWERED NAVAL INSTALLATIONS.

By A. M. Procter, Captain, U. S. N., Retired, Member.

In the following discussion an attempt will be made, from an examination of the evidence available, to determine the possibilities of the two-cycle, double acting engine on the basis of present day practice and knowledge.

It may be stated with certainty that, due to improvements in materials and to increased knowledge, the future will see vast improvements in the heavy-oil engine. With these we are not concerned.

The conclusions reached in this discussion represents what we believe to be possible now.

The results predicted, which are in advance of existing installations, cannot be produced overnight. Time, money and a large amount of development work will be necessary.

There is, however, one outstanding advantage of the Diesel engine over all other existing types of prime movers. The cost of development work, sufficient to justify a radical step beyond present practice, is but a small fraction of the cost of a completed installation.

If, as in the steam turbine plant, improvements can only be made by experiment with a completed installation, the risk of failure makes it necessary to proceed with caution.

If, however, as in the case of the Diesel engine, the experimental features can be tested in one cylinder at a cost representing but a small fraction of that of a completed installation, a very much more radical step in advance is justified. A remarkable example of a successful application of this fact was the development by

Sulzer of a one-cylinder experimental engine, in 1914. At this time the largest cylinder in operation was about 30 inches in diameter. For the purpose of determining the ultimate possible at the time, a cylinder of 39 inches diameter was built and tested at 120 pounds mean indicated pressure. By this bold stroke the whole field was cleared below that point, and results which would have taken years to accomplish, proceeding cautiously inch by inch, were obtained by one comparatively inexpensive experiment.

It detracts nothing from the credit due to the Sulzer organization to record the fact that it was more than ten years before the conservatism of the Diesel engine users was overcome to the point of accepting even a cylinder diameter of 35.5 inches.

The purpose of this study is, primarily, to determine the possibilities of the Diesel engine for Naval purposes. The results are applicable to any field in which the high speed engine is suitable, with the reservation that for continuous operating conditions the powers permissible are about 60 per cent of those for full power Naval conditions.

The conditions of the problem require an engine combining the maximum economy of weight and space, with ability to stand forcing, for short periods, to a power about double that established for continuous operating conditions.

In order to meet these conditions it is necessary to combine a low value of weight per cubic inch cylinder volume, a high value of mean indicated pressure, and a high piston speed, with a high value of R.P.M.

These factors are interdependent and conflicting. In order to get the high piston speed required, the greatest possible rigidity of frame and bedplate is necessary, which does not help in the matter of low weight per cubic inch cylinder volume. As the piston speed increases it becomes more difficult to maintain the scavenging efficiency upon which the mean indicated pressure depends.

THE LAW OF SIMILITUDE.

In the course of this discussion reference will be made to the law of similitude. This law was first stated by Normand in 1896, in the following words: "A drawing of an engine, working at a

fixed pressure, may be used on any scale, without the load on the material, and the rubbing surfaces, or the steam passages being modified; provided the number of revolutions be made inversely proportional to the linear dimensions. The power of the engine under the same conditions as regards inertia strains is then proportional to the square of the linear dimensions, and its weight per H.P. to the linear dimension. I have repeatedly built engines of different sizes from the same drawing by simply changing the scales and the figures."

There are two factors which enter into the problem which require some modification of this law.

The first of these concerns the matter of bearing loads and their effect in limiting piston speeds. This will be discussed at some length later.

The second question is that of the effect of varying casting sections upon the weight per cubic inch cylinder volume. As far as strength is concerned the law of similitude holds — the unit stresses will be the same for all cylinder sizes.

There is however a widespread belief that foundry limitations require relatively larger sections in small engines than in large engines. That this is the practice of marine engine builders is undoubtedly true. That it is necessary, within the limits with which we are concerned, is open to question.

Certain data is shown on Figure 1, which bears upon this point. It will be noted that the relative thickness of the steel castings in engines built for light weight per B.H.P. are about 60 per cent of that for cast-iron engines, for the same cylinder diameters. Various points have been plotted which show nothing beyond the fact that there is a wide difference of opinion between designers and a wide difference between the standards of automotive and marine practice. If the points for the automobile engines are compared with those for much larger marine engines, it is evident that there is little basis for the assumption that it is necessary to make the ratio between casting thickness and cylinder diameter greater in small engines than in large ones. In very small cylinder diameters there is, undoubtedly, a point beyond which we cannot go, but with that we are not concerned, since it will be far below the limits considered in this discussion.

EFFECT OF LAW OF SIMILITUDE UPON FACTORS LIMITING PISTON SPEED.

There are three factors which directly and indirectly limit the piston speed at which an engine may be run. These factors are:

1 — Inertia forces;
2 — Bearing loads;
3 — Scavenging conditions.

Assuming similarity, the effect of varying the cylinder diameters does not change the unit values of the above factors. For the

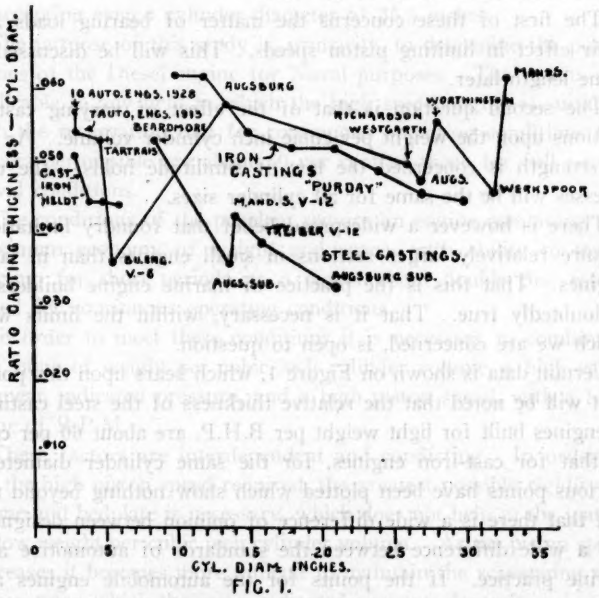


FIG. 1.

same piston speed, the unit stresses due to inertia are the same for all sizes. The bending stresses are the same. The values of unit bearing loads due to inertia and pressure are the same. In regard to scavenging, the velocities through the scavenging ports are independent of the cylinder sizes. In other words, the law of similitude holds for all of the above factors with the modification that,

while the unit bearing loads are the same for all cylinder sizes, there is question as to whether the same unit loads can be carried in large bearings as in small ones.

This will later be discussed in detail.

THE EFFECT OF INERTIA STRESSES.

In high speed engines the effect of inertia stresses upon the engine framing and the structure supporting the engine may be very great. Material deflections in the engine structure may be caused with consequent reaction upon the bearing conditions and other factors. The engine may have a theoretically perfect balance, but the effectiveness of this balance rests on the assumption that the engine framing is infinitely rigid. This condition has been reached for all practical purposes in the block cylinder casting of automobile engines which now run without the slightest evidence of unbalance at piston speeds which would have been considered impossible of attainment a few years ago. In the marine field there is evidence that the necessity for greater rigidity is now appreciated, but it must be realized that owing to the relatively larger weight of reciprocating parts the problem, in the case of a double acting cross head engine, is a far more difficult one than in the case of trunk piston engines of small size.

There is no way in which this problem can be handled except by comparing the forces expected in a proposed engine with those found in engines in successful operation. In the case of two-cycle double acting engines there is very little data available. The highest known piston speed for two-cycle marine engines in actual service is about 1000 feet per minute in the Augustus.

The reciprocating weights are unknown, but an inspection of the drawings of this engine indicates that they will not be materially different from those of the Worthington engine, which have a value of 32 pounds per square inch of piston. For the purpose of comparing the results with those of an engine designated later as design "B," they may be reduced to a 20.0-inch cylinder diameter.

Since in similar engines the weight per square inch piston varies directly as the linear dimension, the corresponding weight will be

about 23 pounds per square inch piston. The resultant conditions for an Augsburg engine similar to that of the Augustus will be as follows:

Cylinder diameter	20.0	inches
Stroke	35.0	inches = 2.92 feet
Crank radius	1.5	feet
Ratio $\frac{\text{stroke}}{\text{bore}}$	1.75	
Weight per square inch piston	23	pounds
R.P.M. for 1000 feet minimum	171	

Maximum inertia force = 438 pounds square inch piston.
$(.00034 \times 23 \times 1.4 \times 171^2 \times 1.25)$

Undoubtedly this engine could be run at a materially higher piston speed before reaching a limit imposed by the value of the inertia forces.

The Fiat engine with about the same piston speed will show materially higher values of the inertia forces due to the shorter ratio of stroke to bore. This follows from the fact that a decrease in the stroke bore ratio results in a negligible decrease in the reciprocating weights, while the revolutions are proportionally increased.

Using a value of 20 pounds per square inch piston, which is conservative, the results for a 20-inch cylinder diameter will be as follows, assuming that this engine normally rated at 1000 feet piston speed can be run at an overload ratio of 1200 feet:

Ratio $\frac{\text{stroke}}{\text{bore}}$	1.2	
Stroke	24	inches
Crank radius	1.0	feet
R.P.M. at 1200 feet minimum piston speed	250	
Maximum inertia force	531	pounds per square inch

$(.00034 \times 20 \times 1 \times 250^2 \times 1.25)$

It is obvious that, from a consideration of the effect of inertia forces upon the engine framing, and as will be shown later, upon the bearing load, the weight of the reciprocating parts must be

reduced to a minimum if really high piston speeds are to be attempted. In order to determine what value is possible, the reciprocating parts, for a two-cycle engine, have been laid down and the weights calculated in some detail.

By making the greatest possible use of lightweight materials, the total weight can be brought down to about 8.5 pounds per square inch piston for a 20-inch cylinder diameter. With a ratio of stroke to bore of 1.5 the results will be as follows for a piston speed of 1800 feet minimum:

Stroke	30.0	inches
Crank radius	1.25	feet
R.P.M.	360	
Maximum inertia force	627	pounds square inch piston

$(.00034 \times 8.5 \times 1.25 \times 360^2 \times 1.25)$

The resultant inertia forces will not be greater than those found in high speed automobile engines, when compared on the basis of the maximum gas pressures, which determine the designed stresses. The results for a well known standard automobile engine are as follows:

Cylinder diameter	3.25	inches
Stroke	4.5	inches
Ratio $\frac{\text{stroke}}{\text{bore}}$	1.38	
Crank radius	.1875	feet
Weight reciprocating parts	.38	pounds square inch piston
R.P.M.	3600	
Piston speed	2700	feet minimum
Maximum inertia force	454	pounds square inch piston

$(.00034 \times .38 \times .187 \times 3600^2 \times 1.25)$

$$\text{Ratio } \frac{\text{inertia pressure}}{\text{gas pressure}} = \frac{454}{375} = 1.21$$

For the design engine the above ratio would give a gas pressure of 520 pounds per square inch, which is not greater than that

usually found in engines running on the Diesel cycle. Without regard to the effect of inertia stresses upon other factors, it is evident that the problem of handling inertia stresses in the framing of two-cycle double acting engines running at 1800 feet per minimum piston speed is one which can be solved by designing a structure having the same relative girder strength found in high speed automotive engines.

EFFECT OF PISTON SPEED ON SCAVENGING EFFICIENCY.

There is no subject connected with the design of two-cycle engines more important than that of scavenging, and there is no subject which has been more casually treated by the writers on the subject of Diesel engine design.

The problem of scavenging in a two-cycle engine is one of great complexity, when the attempt to handle it mathematically is made. If it were a question only of determining the velocity of air through an orifice into a receiver of lower pressure, the problem could be very simply solved by the equation.

$$V = K \sqrt{T_1 \left\{ 1 - \left(\frac{p_o}{p} \right)^n \right\}}$$

A curve showing velocities in feet per second for various values of scavenging air pressure is shown on Figure 2, curve No. 1 for values of $T = 540$ degrees absolute and $p_o = 14.7$ pounds absolute. This curve is of no value except for the purpose of showing the general character of the velocity — air pressure relationship.

The difficulty arises from the fact that when the scavenging ports are opened the air will enter the cylinder at a much more rapid rate than it can be forced out, since there is in the cylinder insufficient pressure to give a velocity through the exhaust ports sufficient to pass out an amount of air equal to that which is entering.

This will naturally build up a pressure in the cylinder and if the process were continuing, a balance would be established and the pressure in the cylinder would be roughly one-half that in the scavenging belt.

The velocity through the scavenging ports would, under these conditions, be something like that represented by curve No. 3.

But the process is not so simple. With the short time interval, the constantly changing relation between the areas of the exhaust ports and the scavenging ports, and the constantly changing value of the pressure in the cylinder, a situation is produced which defies solution if an attempt is made to handle it mathematically. One thing, however, is certain,— the curves shown on Figure 1 give the general character of the velocity curves, and somewhere between the limits shown by curves No. 1 and No. 4, there will be a curve

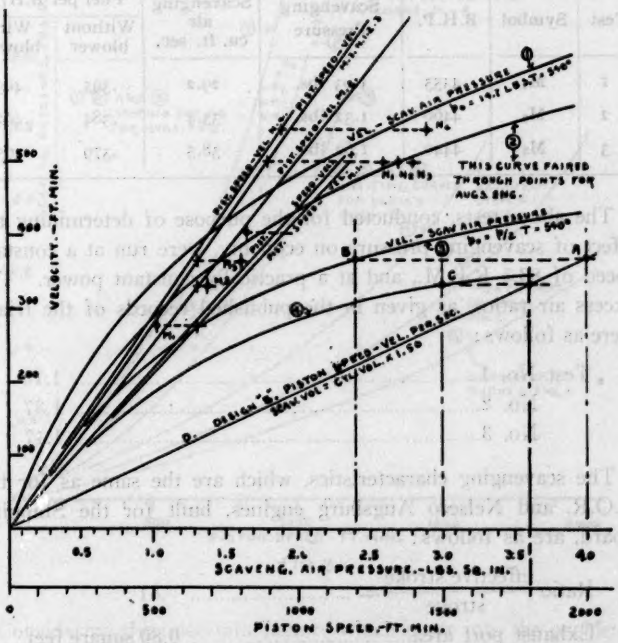


FIG. 2

which represents the relationship existing between velocity and scavenging pressure for any engine under consideration.

With sufficient data from trial results, covering a wide range of speed and power, it would be possible to establish points on the velocity-scavenging air pressure curve, which, when connected up

with the velocity-piston speed curves, would establish the scavenging conditions with a fair amount of accuracy.

A search for data on this subject is not particularly helpful. The best results found are from the trials of the Augsburg two-cycle D.A. engine, which are tabulated below:

Engine: Augsburg, "Magdeburg."

Cylinder diameter 27.6 inches, stroke 47.2 inches.

Test	Symbol	B.H.P.	Scavenging Pressure	Scavenging air cu. ft. sec.	Fuel per B.H.P.	
					Without blower	With blower
1	M_1	4355	1.03 lbs.	29.2	.393	.409
2	M_2	4460	1.32 lbs.	35.4	.384	.402
3	M_3	4440	1.62 lbs.	38.5	.379	.405

The above tests, conducted for the purpose of determining the effect of scavenging pressure on economy, were run at a constant speed of 83.5 R.P.M., and at a practically constant power. The excess air ratios, as given in the published records of the trials, were as follows:

Test No. 1..... 1.15
No. 2..... 1.37
No. 3..... 1.47

The scavenging characteristics, which are the same as for the H.O.R. and Nelseco Augsburg engines, built for the Shipping Board, are as follows:

Ratio $\frac{\text{effective stroke}}{\text{stroke}} =$ ..... .81
Exhaust port area..... 0.80 square feet
Scavenging port area..... 0.71 square feet

Using the above data, the velocity through scavenging ports has been computed for the piston speed, which, in all cases, is 656 feet per minute, and the velocity curves M_1, M_2 and M_3 drawn. By connecting up the points on these curves with the piston speed the points M_1, M_2 and M_3 are established.

By using similar data for the Nelseco engine, and for the Hamburg 15,000 engine, the points H and N_1, N_2, N_3 and N_4 are established. Using these points, the curve No. 3 has been drawn. In Figure 3 the data from these engines have been plotted on a piston speed and scavenging pressure basis, and limiting curves Nos. 1 and 3 drawn.

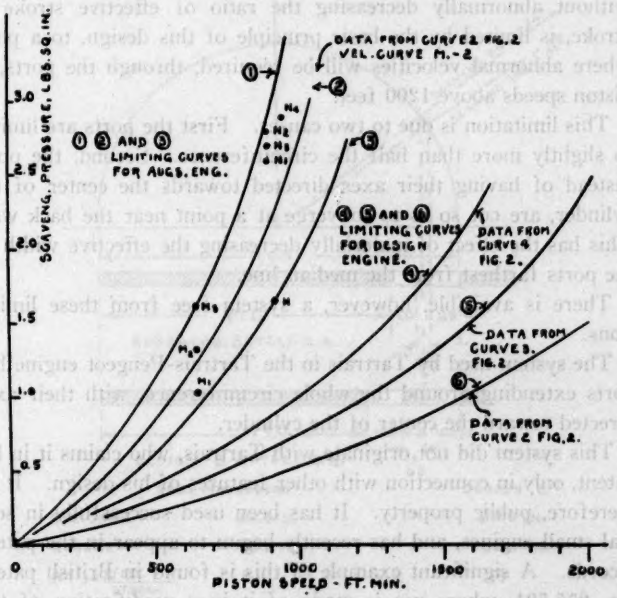


FIG. 3.

Considering the uncertain factors which enter into the problem, it is probable that these two curves, one drawn through the point for the Hamburg engine, and the other drawn through the point M_3, represent the limits between which the scavenging air pressure values fall, for various piston speeds.

An examination of these curves indicates one fact of importance — that with this engine a limiting piston speed of about 1200 feet per minute must be accepted, with 3.0 pounds scavenging pressure,

which is the maximum which can be reached with a single stage scavenging blower.

The scavenging system used on this engine has shown remarkably efficient results. When compared with other engines a high scavenging efficiency is indicated with moderate scavenging pressures. There is, however, one limitation to its use for high piston speeds. The area of the scavenging and exhaust ports possible, without abnormally decreasing the ratio of effective stroke to stroke, is limited by the basic principle of this design, to a point where abnormal velocities will be required, through the ports, at piston speeds above 1200 feet.

This limitation is due to two causes. First the ports are limited to slightly more than half the circumference. Second, the ports instead of having their axes directed towards the center of the cylinder, are cut so as to converge at a point near the back wall. This has the effect of materially decreasing the effective width of the ports farthest from the median line.

There is available, however, a system free from these limitations.

The system used by Tartrais in the Tartrais-Pengeot engine has ports extending around the whole circumference, with their axes directed toward the center of the cylinder.

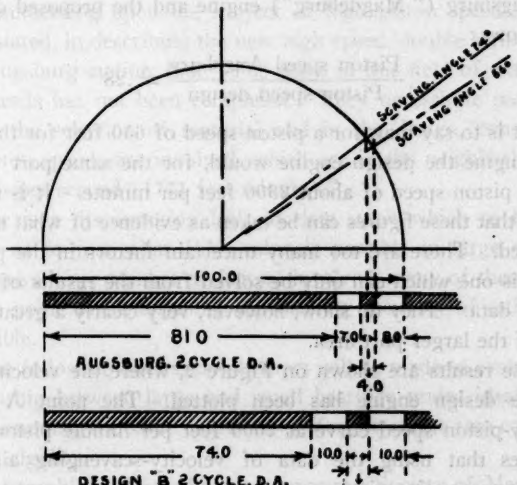
This system did not originate with Tartrais, who claims it in his patent, only in connection with other features of his design. It is, therefore, public property. It has been used successfully in several small engines, and has recently begun to appear in the patent records. A significant example of this is found in British patent No. 275,591, where use is made of it in a modification of the Sulzer system.

The fact that the system used by Tartrais is an efficient one is indicated by the results in the Pengeot engine, where a brake M.E.P. of 70.5 pounds is developed at 1430 feet piston speed, with a relatively high ratio of effective stroke to stroke,— 78 per cent,— and an attached scavenging pump. With 75 per cent mechanical efficiency the mean indicated pressure would be 94 pounds per square inch.

Figure 3a shows the characteristics of the Augsburg engine compared with a design for an engine, using the Tartrais system,

with a ratio of effective stroke to stroke equal to 74 per cent. The relatively large port area possible is due to causes previously noted.

For the same piston speed, and the same relative scavenging volume, the velocity through the scavenging ports will be three and one-half times as great, in Augsburg engine, as in the engine proposed.



	AUGS. "MAGDEBURG"	DESIGN "B"
CYL. DIAM.	27.6 IN.	20.0 IN.
STROKE	47.2 IN.	30.0 IN.
RATIO $\frac{\text{EFFEC. STR.}}{\text{STROKE}}$	.81	.74
SCAVENG PORT AREA	.71 SQ. FT.	1.00 SQ. FT.
SCAVENG. ANGLE	66°	74°
SCAVENG. ANGLE / 360°	.183	.205
CYL. VOL. (NET) PER STR.	14.7 CU. FT.	5.2 CU. FT.

$$\text{VELOCITY THROUGH PORTS} = \frac{\text{VOL} \times \text{R.P.M.} \times 360}{\text{SCAVENG. ANGLE} \times \text{PORT AREA} \times 60 \times K} \qquad K = .66$$

$$\text{RELATIVE VELOCITY AT SAME PISTON SPEED} = \frac{\text{VOL.}}{\text{SCAV. ANGLE} \times \text{STROKE} \times \text{PORT AREA}}$$

RELATIVE VEL. 3.5 : 1.0

FIG. 3a.

This may be expressed another way. For the same velocity through the ports, the relative piston speeds are represented by the expression

$$\frac{\text{Piston speed}_1}{\text{Piston speed}_2} = \frac{\text{vol.}_1 \times \text{scav. angle}_1 \times \text{port area}_1 \times \text{stroke}_1}{\text{vol.}_2 \times \text{scav. angle}_2 \times \text{port area}_2 \times \text{stroke}_2}$$

Using the values shown on Figure 3a, the relationship between the Augsburg ("Magdeburg") engine and the proposed design is as follows:

$$\frac{\text{Piston speed Augsburg}}{\text{Piston speed design}} = .28$$

That is to say that, for a piston speed of 650 feet for the Augsburg engine the design engine would, for the same port velocity, have a piston speed of about 2300 feet per minute. It is not suggested that these figures can be taken as evidence of what might be expected. There are too many uncertain factors in the problem, which is one which can only be solved from the results of experimental data. They do show, however, very clearly a great advantage of the larger port area.

These results are shown on Figure 2, where the velocity curve for the design engine has been plotted. The point A on the velocity-piston speed curve at 2000 feet per minute piston speed, indicates that using the data of velocity-scavenging air curve No. 3, a scavenging air pressure of 2.4 pounds per square inch will be sufficient.

Considering all of the uncertainties of the problem, the indications are that a scavenging pressure of 3 pounds per square inch will give an ample margin, for piston speeds up to 1800 feet per minute, to maintain the same scavenging conditions existing in the Augsburg engine at its rated power. This is indicated by Figure 3, to which the results from Figure 2 have been transferred.

EFFECT OF BEARING LOAD IN LIMITING PISTON SPEED.

In the development of the aero engine there has been a progressive increase in bearing loads. A captured Maybach engine was tested during the war, and the results for this engine published about 1919, by the British Air Ministry, showed a bearing load of 9000 foot pounds per second.

This factor, called P.V. for short, has increased since that time to 53,000, in the Packard X-24 engine, which runs at the unprecedented value of rubbing velocity of 83 feet per second.

There can be no doubt but that with perfect balance, adequate forced lubrication, and the nearly perfect rigidity found in the block cylinder construction of the automotive engine, a large increase in the P.V. value would be possible in the marine engine.

In commenting upon the subject of high piston speeds, P. V. Schuh stated, in describing the new high speed, double acting, two-cycle Augsburg engine, that "The work in this field of extremely high speeds has not been completed. They necessitate particular care in the selection of material and in the design, considering that piston speeds are involved, which are in the neighborhood of 9 metres per second (1771 feet minute)."

There is a discouraging lack of data upon which to base an opinion as to just what P.V. value is possible in large engines. There are a large number of factors involved, many of them variable to a point where anything approaching accuracy of estimate is impossible.

In the following discussion attempts will be made to evaluate the relationship between large and small bearings under ideal conditions, assuming the same degree of perfection in the two cases and perfect rigidity.

In a pamphlet published by the American Society of Mechanical Engineers, under the title "A Graphical Analysis of Journal Lubrication," by H. A. S. Howarth, the following formula, developed by Harrison, is given:

$$R = \frac{12\pi M U a^2}{n^2} \left\{ \frac{c}{(2+c^2)(1-c^2)^{\frac{1}{2}}} \right\}$$

R = Load per unit length.
M = Absolute viscosity.
U = Rubbing velocity.
a = Radius of journal.
n = Radial clearance.

If the same values of M, U and a/n are assumed, the value of R for various sized bearings will be given by the bracketed mem-

ber of the above formula. The value of C is shown in Figure 4a. This value of C depends upon the minimum oil thickness. The greater the pressure the smaller the oil thickness and the greater the value of C, which reaches a value of unity when metallic contact is established.

The actual value of the thickness of the oil film, before breakdown occurs, is doubtful, but since we are concerned with ratios and not absolute values it is not important. It may be assumed that with an equal degree of absolute perfection of machining and fitting, and with no deflection, this value will be the same in all bearings.

Curves for the relative value of

$$\frac{C}{(2 + c^2)\,(1 - c^2)^{\frac{1}{2}}}$$

which give the relative values of R for various sizes of shafts, have been plotted on Figure 4. These curves are indicated as numbers 1, 2 and 3.

Curves numbers 4, 5 and 6 show the relative unit bearing load for similar bearings, having the same ratio of length to diameter. The three curves 4, 5 and 6 are of the general character expressed by the relationship $PV \times \text{shaft diameter}^n = \text{constant}$

The value $n = .5$ may be taken as a close approximation.

The results shown on Figure 4 follow from certain assumptions, which are far from being correct when applied to a practical rather than an ideal case.

It was assumed that M, the coefficient of viscosity, was the same for all bearings.

The result of assuming a larger value of M in larger bearings will be to increase the relative load which the large bearings will carry.

On the other hand, the variations from perfection, the variations due to imperfect alignment, and the variations due to deflection, being proportional to the linear dimensions, will be greater in proportion to the oil film thickness, as the size of the bearing increases and will tend to decrease the value of the load which can be carried in the larger bearings.

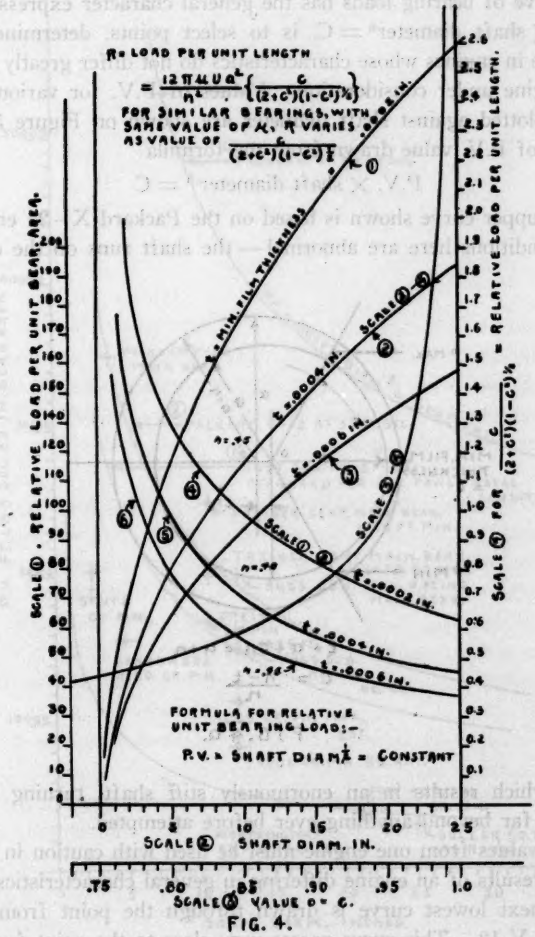

R = LOAD PER UNIT LENGTH
$= \frac{12\pi\mu U a^2}{h^2} \cdot \left\{\frac{c}{(2+c^2)(1-c^2)^{\frac{1}{2}}}\right\}$
FOR SIMILAR BEARINGS, WITH SAME VALUE OF μ, R VARIES AS VALUE OF $\frac{c}{(2+c^2)(1-c^2)^{\frac{1}{2}}}$
h = MIN. FILM THICKNESS
h = .0008 IN.
h = .0006 IN.
h = .0004 IN.
h = .0002 IN.
n = .95
n = .90
SCALE ① - ③
SCALE ① - ④
SCALE ① - ⑤
SCALE ① - ⑥
SCALE ① - RELATIVE LOAD PER UNIT BEAR. AREA.
SCALE ③ FOR $\frac{c}{(2+c^2)(1-c^2)^{\frac{1}{2}}}$ = RELATIVE LOAD PER UNIT LENGTH.
FORMULA FOR RELATIVE UNIT BEARING LOAD:-
P.V. ÷ SHAFT DIAM.² = CONSTANT
SCALE ② SHAFT DIAM. IN.
SCALE ④ VALUE OF c.

FIG. 4.

There is at present insufficient data even to attempt to evaluate these factors. The best that can be done, recognizing the fact that the curve of bearing loads has the general character expressed by P.V. $\times$ shaft diameter^n = C, is to select points, determined by practice in engines whose characteristics do not differ greatly from the engine under consideration. Values of P.V. for various engines plotted against shaft diameter are shown on Figure 5 and curves of P.V. value drawn from the formula

$$\text{P.V.} \times \text{shaft diameter}^{.5} = \text{C}$$

The upper curve shown is based on the Packard X – 24 engine. The conditions here are abnormal — the shaft runs on the crank

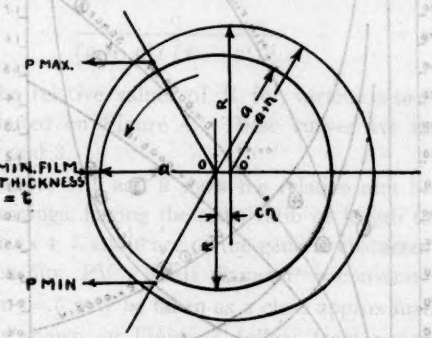


FIG. 4a.

webs, which results in an enormously stiff shaft, turning at a velocity far beyond anything ever before attempted.

P.V. values from one engine must be used with caution in predicting results of an engine differing in general characteristics.

The next lowest curve is drawn through the point from the Packard V-12. This curve passes very close to the point for the center main bearing of the Augsburg submarine engine at maximum rating, which carries the heaviest load in its class of any engine of which there is any authentic record.

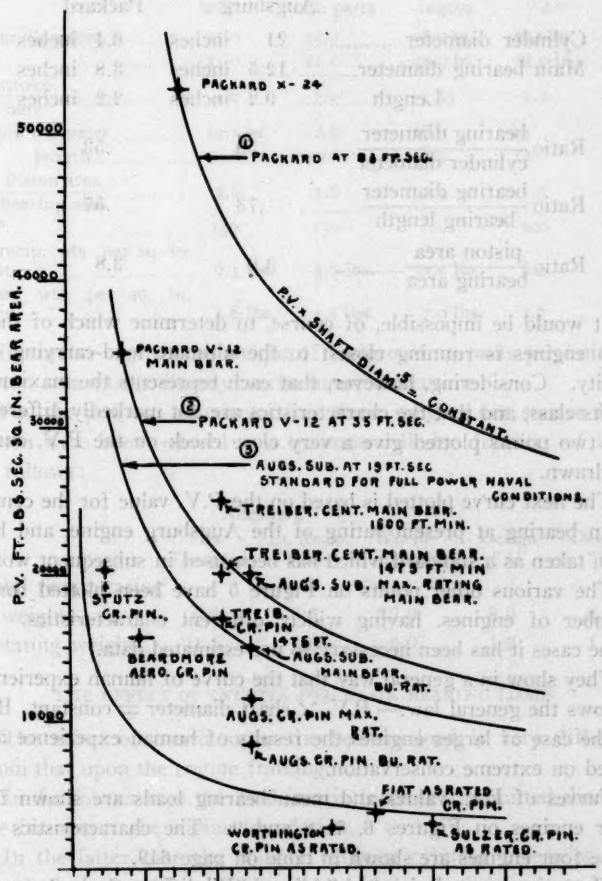

PACKARD X-24
50000
① PACKARD AT 88 FT. SEC.
40000
PACKARD V-12 MAIN BEAR.
② PACKARD V-12 AT 35 FT. SEC.
30000
③ AUGS. SUB. AT 19 FT. SEC STANDARD FOR FULL POWER NAVAL CONDITIONS.
TREIBER. CENT. MAIN BEAR. 1600 FT. MIN.
TREIBER. CENT. MAIN BEAR. 1475 FT. MIN
AUGS. SUB. MAX. RATING MAIN BEAR.
STUTZ CR. PIN.
TREIB. CR. PIN 1475 FT.
BEARDMORE AERO. CR. PIN
AUGS. SUB. MAIN BEAR. BU. RAT.
AUGS. CR. PIN MAX. RAT.
AUGS. CR. PIN. BU. RAT.
FIAT AS RATED CR. PIN.
WORTHINGTON CR. PIN AS RATED.
SULZER. CR. PIN. AS RATED.
P.V. FT. LBS. SEC. SQ. IN. BEAR AREA.
PV × SHAFT DIAM. S = CONSTANT.
5
10
15
20
25
30
SHAFT DIAM. INCHES.

FIG. 5.

The approximate shaft characteristics of these two engines are as follows:

	Augsburg	Packard
Cylinder diameter	21 inches	6.4 inches
Main bearing diameter	12.5 inches	3.8 inches
Length	9.2 inches	2.2 inches
Ratio $\frac{\text{bearing diameter}}{\text{cylinder diameter}}$	.59	.53
Ratio $\frac{\text{bearing diameter}}{\text{bearing length}}$	.73	.57
Ratio $\frac{\text{piston area}}{\text{bearing area}}$	3.0	3.8

It would be impossible, of course, to determine which of these two engines is running closest to the ultimate load-carrying capacity. Considering, however, that each represents the maximum of its class, and that the characteristics are not markedly different, the two points plotted give a very close check on the P.V. curve as drawn.

The next curve plotted is based on the P.V. value for the center main bearing at present rating of the Augsburg engine, and has been taken as a standard which has been used in subsequent work.

The various other points on Figure 5 have been plotted for a number of engines, having widely different characteristics. In some cases it has been necessary to use estimated data.

They show in a general way that the curve of human experience follows the general law,— P.V. $\times$ shaft diameter = constant. But in the case of larger engines the results of human experience are based on extreme conservation.

Curves of P.V. values and mean bearing loads are shown for four engines on Figures 6, 7, 8 and 9. The characteristics of these four engines are shown in table on page 649.

The two engines designated "A" and "B" differ only in the ratio of stroke to bore. Lengthening the ratio of stroke to bore results in a negligible increase in the weights of reciprocating parts.

These weights have been estimated in some detail and represent the minimum which can be obtained by careful design, and the use of light materials.

	Augsburg Sub.				
	As built	Light recip. parts	Worthington	Design "A"	Design "B"
Cylinder diameter	21.0	21.0	28.0 in.	20.0 in.	20.0 in.
Stroke	21.0	21.0	40.0 in.	24.0 in.	30.0 in.
Ratio $\frac{\text{stroke}}{\text{bore}}$	1.0	1.0	1.43	1.2	1.5
Crankpin diameter	12.5 in.	12.5	18.0	14	14
length	10.6	10.6	18.0	12	12
Ratio $\frac{\text{Piston area}}{\text{Bearing area}}$	2.6	2.6	1.9	1.9	1.9
M. I. P.	130	130	100	100	100
Total recip. wts. per sq. in. piston	6.3 lbs.	3.8 lbs.	32.0 lbs.	8.0	8.4
Rotating wts. per sq. in. piston	1.6 lbs.	1.1 lbs.	7.0 lbs.	1.7	1.7

In similar engines the weight of reciprocating parts per unit of piston area varies directly as the linear dimension.

For the purpose of comparison, these weights per square inch piston have been reduced to a common cylinder diameter of 20.0 as follows:

	Augsburg				
	As Designed	Lt. Recip. Parts	Worthington	Design "A"	Design "B"
Total reciprocating weights	6.0	3.6	23.0	8.0	8.4
Rotating weights	1.5	1.0	5.0	1.7	1.7

THE EFFECT OF INERTIA FORCES ON BEARING LOADS.

The effect of inertia upon the bearing load is very different from that upon the engine framing.

In the former case the total inertia force is transmitted through the framing to the engine foundation.

In the latter case the balance between cylinder pressures and inertia forces modifies to a very material degree the resultant bearing load.

In a four-cycle, single acting engine the effect during the four strokes is as follows:

Power Stroke: If the maximum inertia stress is less than the pressure stress, there is no change in the value of the mean pres-

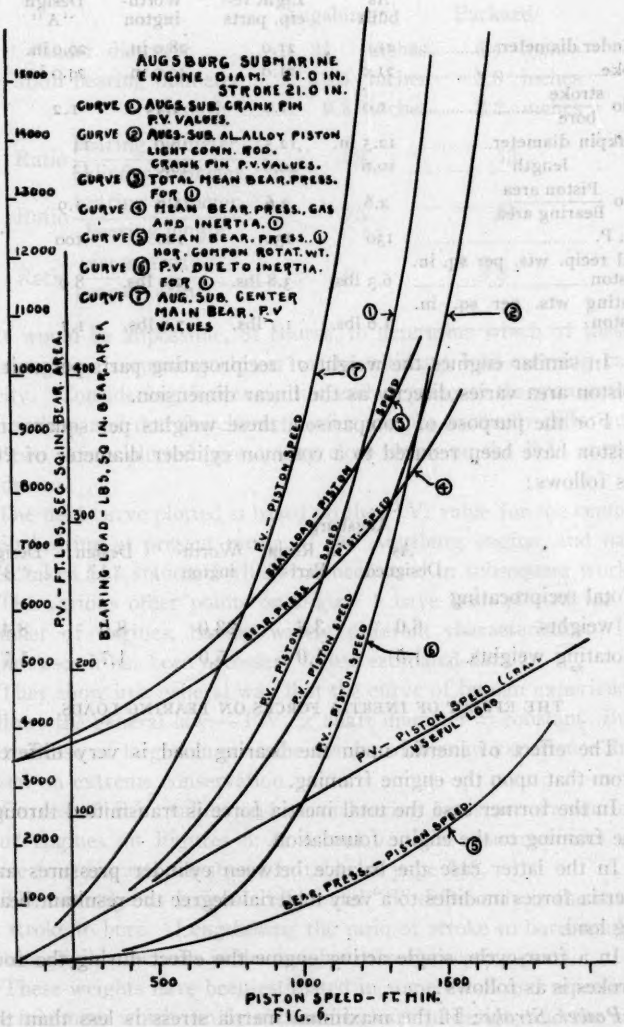

AUGSBURG SUBMARINE
ENGINE DIAM. 21.0 IN.
STROKE 21.0 IN.
CURVE ① AUGS. SUB. CRANK PIN P.V. VALUES.
CURVE ② AUGS. SUB. AL. ALLOY PISTON LIGHT COMN. ROD. CRANK PIN P.V. VALUES.
CURVE ③ TOTAL MEAN BEAR. PRESS. FOR ①
CURVE ④ MEAN BEAR. PRESS., GAS AND INERTIA. ①
CURVE ⑤ MEAN BEAR. PRESS. ① HOR. COMPON ROTAT. WT.
CURVE ⑥ P.V. DUE TO INERTIA. FOR ①
CURVE ⑦ AUG. SUB. CENTER MAIN BEAR. P.V. VALUES
P.V. - FT. LBS. SEC. SQ. IN. BEAR. AREA.
BEARING LOAD - LBS. SQ. IN. BEAR. AREA.
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
400
300
200
100
P.V. - PISTON SPEED
BEAR. LOAD - PISTON SPEED
BEAR. PRESS. - PISTON SPEED
P.V. - PISTON SPEED
P.V. - PISTON SPEED
P.V. - PISTON SPEED (LOAD USEFUL LOAD)
BEAR. PRESS. - PISTON SPEED
500
1000
1500
PISTON SPEED - FT. MIN.

FIG. 6.

sure, and the pressure on the bearings is the mean of the gas pressure.

Compression Stroke: The balance between inertia pressures and gas pressures materially decreases the resultant bearing load.

Admission and Exhaust Strokes: There being no appreciable gas pressures, the bearing load is that due to the mean value of the inertia forces.

The effect of considering the mean of the four strokes is that there is a very material increase in the P.V. values as the speed of revolution is increased. This is shown on Figure 6 where curves for the Augsburg 21-inch submarine engine are plotted.

The straight line is the curve of P.V. for the crank pin based only on gas pressures. The actual P.V. curve starts at zero, tangent to the straight line curve, and shows a very rapid rise as the piston speed is increased.

The effect of lightening the reciprocating weights is shown by comparing curve 2 with curve 1. The weights used in plotting curve 2 are the same, based on cylinder volume, as those of an actual engine having an oil cooled aluminum alloy piston and a light connecting rod.

Curve No. 6 shows very clearly the effect of inertia. This curve has been plotted by subtracting the values of the straight line P.V. curve from those of curve 1.

The values of P.V. due to inertia become greater than those due to useful load at about 1000 feet piston speed. At 1400 feet piston speed the value due to inertia is about two and one-quarter times that due to useful load.

Curve No. 7 shows P.V. values for the center main bearing, which carries the heaviest load of all the bearings. This bearing has values about double those for the crank pin.

This is due, primarily, to the fact that the shaft, having no counter weights, the centrifugal force of two crank webs and the crank pin is added to forces due to other causes.

THE EFFECT OF INERTIA FORCES IN TWO-CYCLE, DOUBLE ACTING ENGINES.

Figures 7, 8 and 9 are plotted from the results shown on Figures 10 and 11.

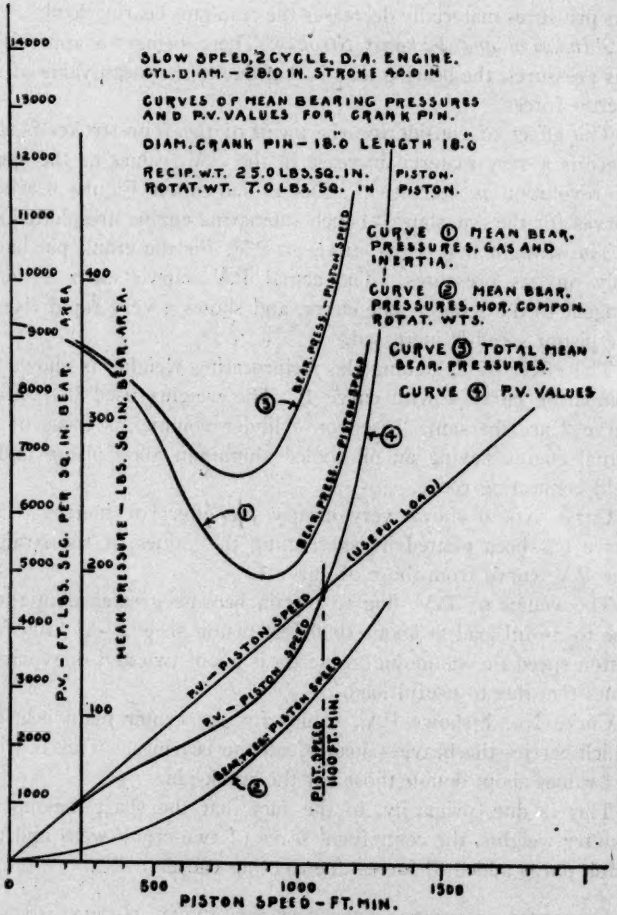

SLOW SPEED, 2 CYCLE, D.A. ENGINE.
CYL. DIAM. – 28.0 IN. STROKE 40.0 IN
CURVES OF MEAN BEARING PRESSURES AND P.V. VALUES FOR CRANK PIN.
DIAM. CRANK PIN – 18.0" LENGTH 18.0"
RECIP. W.T. 25.0 LBS. SQ. IN. PISTON.
ROTAT. WT. 7.0 LBS. SQ. IN PISTON.
CURVE ① MEAN BEAR. PRESSURES, GAS AND INERTIA.
CURVE ② MEAN BEAR. PRESSURES. HOR. COMPON. ROTAT. WTS.
CURVE ③ TOTAL MEAN BEAR. PRESSURES.
CURVE ④ P.V. VALUES
BEAR. PRESS. – PISTON SPEED
(USE FULL LOAD)
P.V. – PISTON SPEED
BEARPRES. – PISTON SPEED
PIST. SPEED 1100 FT. MIN.
P.V. – FT. LBS. SEC. PER SQ. IN. BEAR. AREA
MEAN PRESSURE – LBS. SQ. IN. BEAR. AREA.
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
400
300
200
100
0
500
1000
1500
PISTON SPEED – FT. MIN.

FIG. 7.

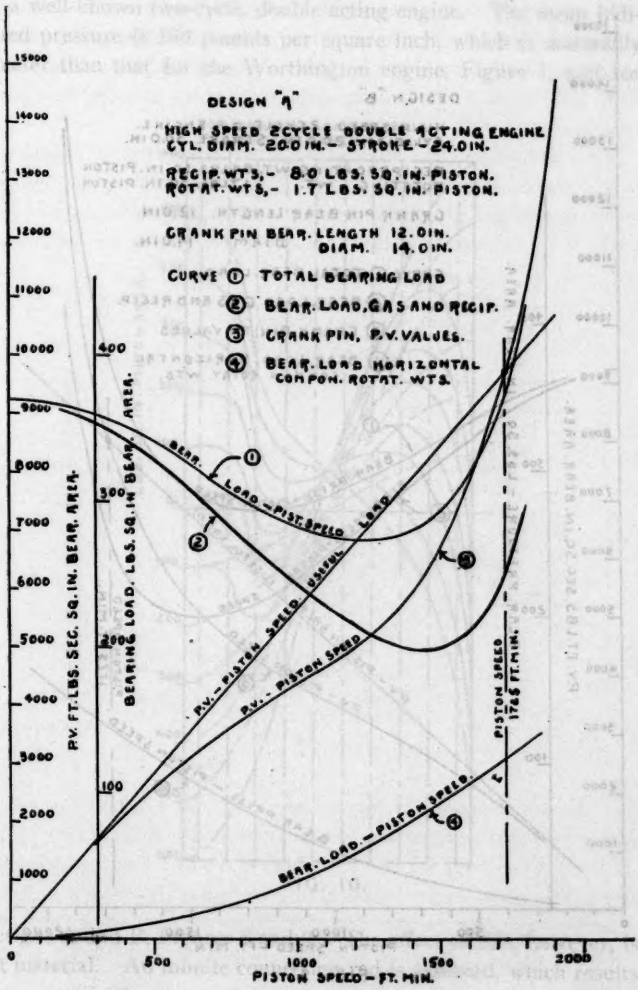

DESIGN "A"
HIGH SPEED 2CYCLE DOUBLE ACTING ENGINE
CYL. DIAM. 20.0 IN. - STROKE - 24.0 IN.
RECIP. WTS. - 8.0 LBS. SQ. IN. PISTON.
ROTAT. WTS. - 1.7 LBS. SQ. IN. PISTON.
CRANK PIN BEAR. LENGTH 12.0 IN.
DIAM. 14.0 IN.
CURVE ① TOTAL BEARING LOAD
② BEAR. LOAD, GAS AND RECIP.
③ CRANK PIN, P.V. VALUES.
④ BEAR. LOAD HORIZONTAL COMPON. ROTAT. WTS.
P.V. FT. LBS. SEC. SQ. IN. BEAR. AREA.
BEARING LOAD. LBS. SQ. IN. BEAR. AREA.
BEAR. LOAD - PIST. SPEED
P.V. - PISTON SPEED
P.V. - PISTON SPEED
BEAR. LOAD - PISTON SPEED
PISTON SPEED 1746 FT. MIN.
PISTON SPEED - FT. MIN.

FIG. 8.

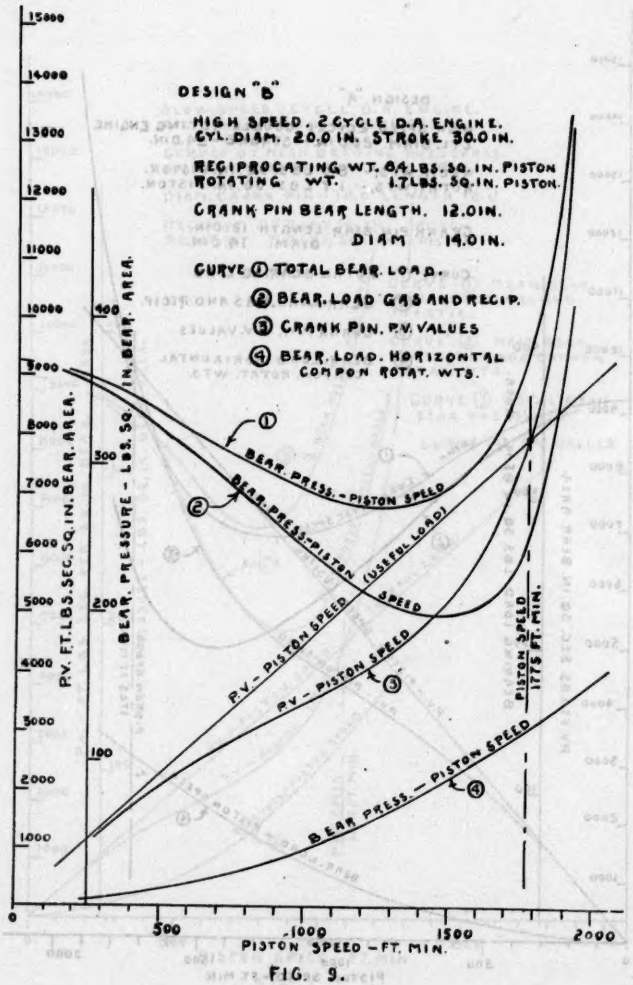

DESIGN "B"
HIGH SPEED, 2 CYCLE D.A. ENGINE.
CYL. DIAM. 20.0 IN. STROKE 30.0 IN.
RECIPROCATING WT. 64 LBS. SQ. IN. PISTON
ROTATING WT. 17 LBS. SQ. IN. PISTON.
CRANK PIN BEAR LENGTH. 12.0 IN.
DIAM 14.0 IN.
CURVE ① TOTAL BEAR. LOAD.
② BEAR. LOAD GAS AND RECIP.
③ CRANK PIN. P.V. VALUES
④ BEAR. LOAD. HORIZONTAL COMPON ROTAT. WTS.
P.V. FT. LBS. SEC. SQ. IN. BEAR. AREA.
BEAR. PRESSURE - LBS. SQ. IN. BEAR. AREA.
BEAR PRESS. - PISTON SPEED
BEAR. PRESS. - PISTON (USEFUL LOAD) SPEED
P.V. - PISTON SPEED
P.V. - PISTON SPEED
PISTON SPEED 1775 FT. MIN.
BEAR PRESS. - PISTON SPEED
PISTON SPEED - FT. MIN.

FIG. 9.

The indicator card used in plotting curves on Figure 10 is that of a well-known two-cycle, double acting engine. The mean indicated pressure is 100 pounds per square inch, which is materially greater than that for the Worthington engine, Figure 7, and for

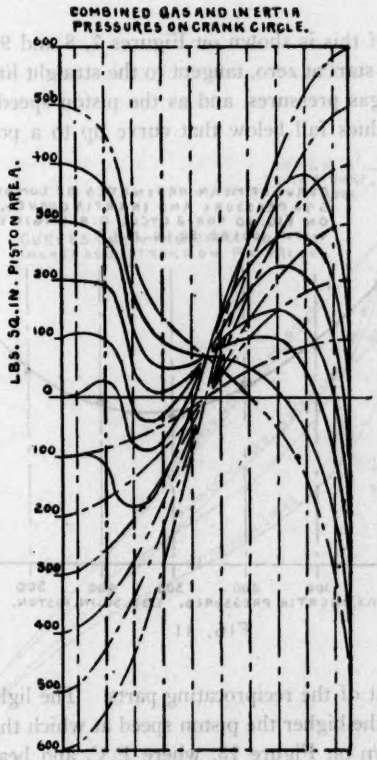


FIG. 10.

Designs A and B, Figures 8 and 9. The effect of this, however, is not material. An infinite connecting rod is assumed, which results in no error in the mean values.

An important fact is disclosed by Figure 11, where the mean ordinates of combined curves of Figure 10 are plotted.

As the inertia values are increased, due either to higher speed or to heavier reciprocating parts, there is a progressive decrease in the resultant piston pressures up to a point where the inertia force has a value of about 350 pounds per square inch piston area. After that point is reached the resultant values increase at a progressive rate.

The effect of this is shown on Figures 7, 8 and 9. The curves of P.V. values start at zero, tangent to the straight line P.V. curves due to useful gas pressures, and as the piston speed increases the actual P.V. values fall below that curve up to a point dependent

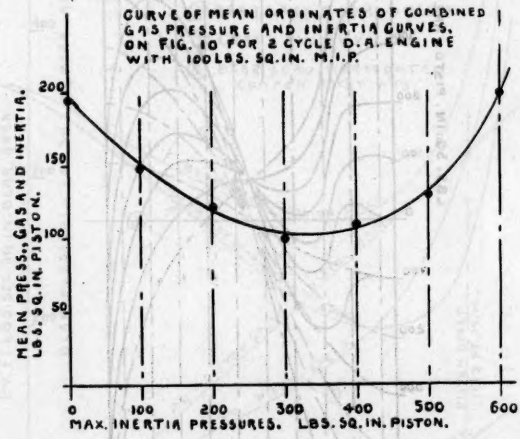


FIG. 11

upon the weight of the reciprocating parts. The lighter the reciprocating parts the higher the piston speed at which this takes place.

This is shown on Figure 13, where P.V. and bearing pressure curves, for a two-cycle double acting engine with varying weights of reciprocating parts, are plotted. These weights vary from the minimum possible up to that found in slow speed double acting engines.

One interesting fact is disclosed by these curves. At 1000 feet piston speed there is no appreciable difference between the results

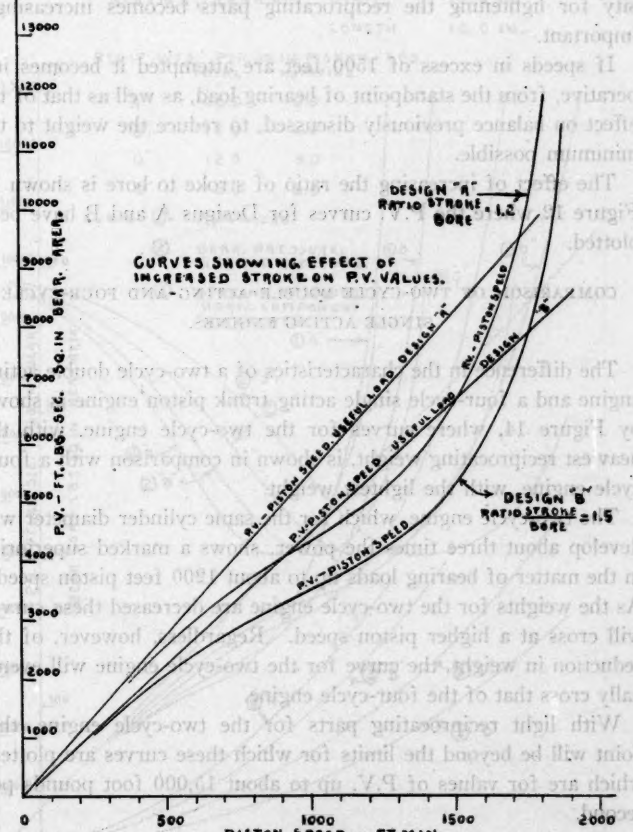

13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
P.V. - FT. LBS. SEC. SQ. IN BEAR. AREA.
0
500
1000
1500
2000
PISTON SPEED - FT. MIN.
DESIGN "A"
RATIO STROKE / BORE = 1.2
CURVES SHOWING EFFECT OF INCREASED STROKE ON P.V. VALUES.
P.V. - PISTON SPEED USEFUL LOAD DESIGN "A"
P.V. - PISTON SPEED
P.V. - PISTON SPEED USEFUL LOAD
DESIGN "B"
DESIGN "B"
RATIO STROKE / BORE = 1.5
P.V. - PISTON SPEED

FIG. 12

from the engine with the heaviest weights and that with the lightest weights. Below 1000 feet piston speed there is actually an advantage in favor of the heavy weights.

As the speed increases above 1000 feet piston speed, the necessity for lightening the reciprocating parts becomes increasingly important.

If speeds in excess of 1500 feet are attempted it becomes imperative, from the standpoint of bearing load, as well as that of the effect on balance previously discussed, to reduce the weight to the minimum possible.

The effect of increasing the ratio of stroke to bore is shown by Figure 12 where the P.V. curves for Designs A and B have been plotted.

COMPARISON OF TWO-CYCLE DOUBLE ACTING AND FOUR-CYCLE SINGLE ACTING ENGINES.

The difference in the characteristics of a two-cycle double acting engine and a four-cycle single acting trunk piston engine is shown by Figure 14, where curves for the two-cycle engine, with the heaviest reciprocating weight, is shown in comparison with a four-cycle engine, with the lightest weight.

The two-cycle engine, which for the same cylinder diameter will develop about three times the power, shows a marked superiority in the matter of bearing loads up to about 1200 feet piston speeds. As the weights for the two-cycle engine are decreased these curves will cross at a higher piston speed. Regardless, however, of the reduction in weight, the curve for the two-cycle engine will eventually cross that of the four-cycle engine.

With light reciprocating parts for the two-cycle engine, this point will be beyond the limits for which these curves are plotted, which are for values of P.V. up to about 15,000 foot pounds per second.

Indications from Figure 15 are that the four-cycle engine will show an advantage for cylinder diameters below 12 inches, from which the inference is clear, that for the field of very high speed lightweight engines of small cylinder diameter, the four-cycle single acting trunk piston engine will remain supreme in its present

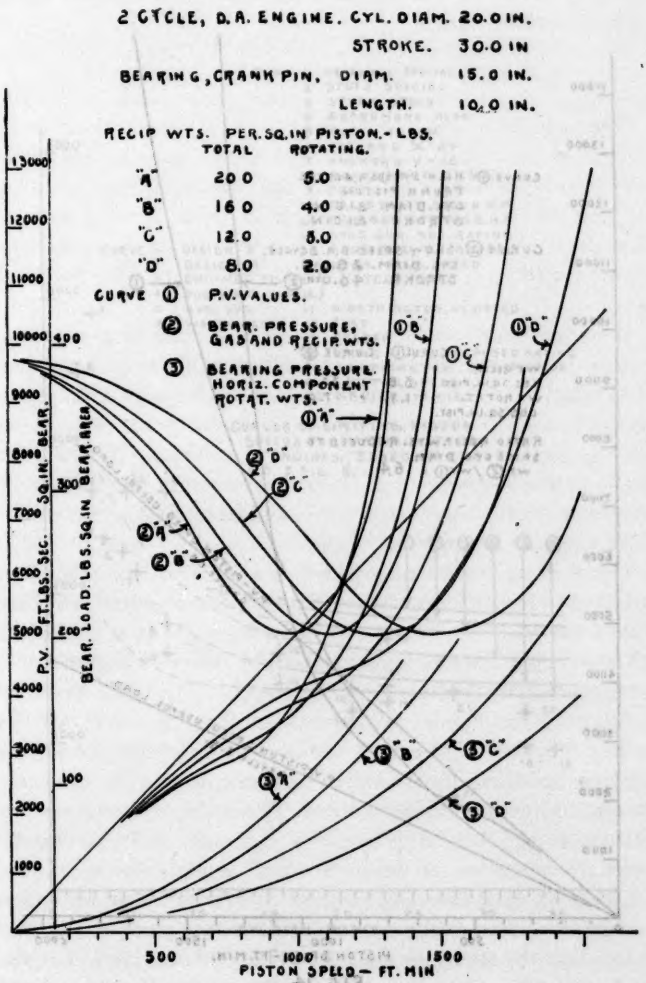

2 CYCLE, D.A. ENGINE. CYL. DIAM. 20.0 IN.
STROKE. 30.0 IN
BEARING, CRANK PIN. DIAM. 15.0 IN.
LENGTH. 10.0 IN.
RECIP WTS. PER SQ IN PISTON. - LBS.
TOTAL ROTATING.
"A" 20.0 5.0
"B" 16.0 4.0
"C" 12.0 3.0
"D" 8.0 2.0
CURVE ① P.V. VALUES.
② BEAR. PRESSURE, GAS AND RECIP WTS.
③ BEARING PRESSURE. HORIZ. COMPONENT ROTAT. WTS.
P.V. FT. LBS. SEC.
BEAR. LOAD. LBS. SQ. IN. BEAR. AREA
SQ. IN. BEAR.
PISTON SPEED — FT. MIN
500
1000
1500

FIG. 13.

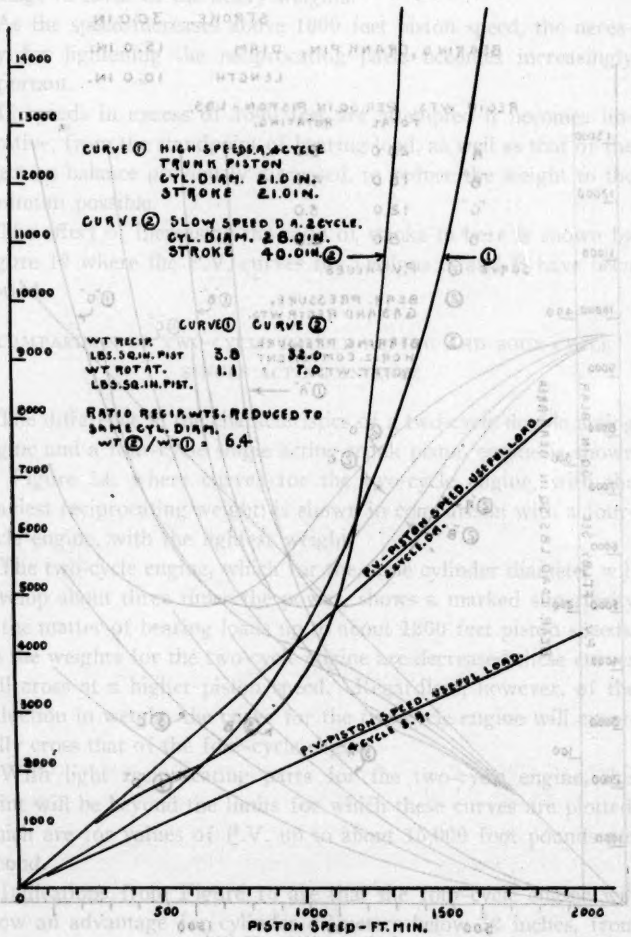

14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
500
1000
1500
2000
PISTON SPEED - FT. MIN.
CURVE ① HIGH SPEED S.A. 4 CYCLE TRUNK PISTON CYL. DIAM. 21.0 IN. STROKE 21.0 IN.
CURVE ② SLOW SPEED D.A. 2 CYCLE. CYL. DIAM. 28.0 IN. STROKE 40.0 IN. ②
①
CURVE ① CURVE ②
WT RECIP. LBS. SQ. IN. PIST 3.8 32.0
WT ROT AT. LBS. SQ. IN. PIST. 1.1 7.0
RATIO RECIP. WTS. REDUCED TO SAME CYL. DIAM. WT ② / WT ① = 6.4
P.V. PISTON SPEED. USEFUL LOAD. 2 CYCLE D.A.
P.V. PISTON SPEED. USEFUL LOAD. 4 CYCLE S.A.

FIG. 14.

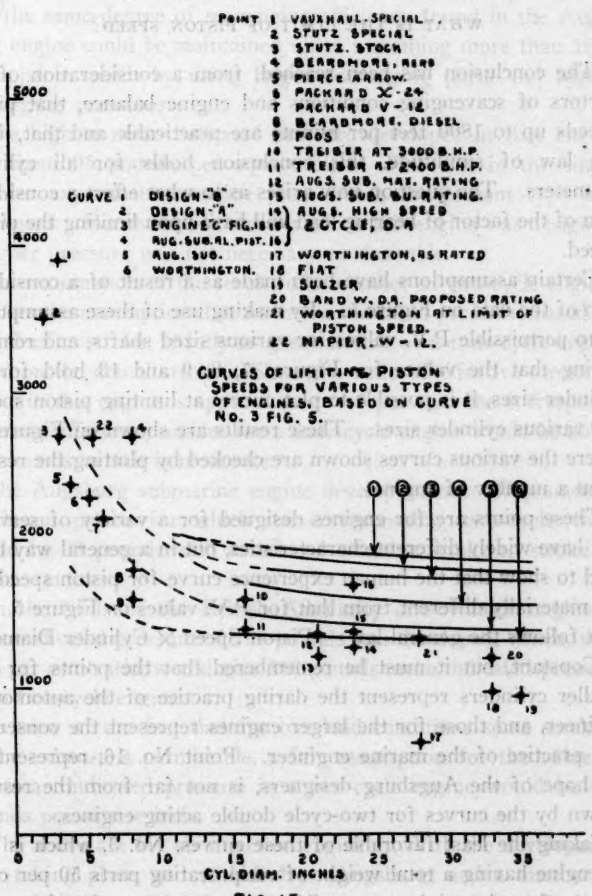

POINT 1 VAUXHAUL-SPECIAL
2 STUTZ SPECIAL
3 STUTZ. STOCK
4 BEARDMORE, NERS
5 PIERCE ARROW.
6 PACKARD X-24
7 PACKARD V-12
8 BEARDMORE. DIESEL
9 FOOS
10 TREIBER AT 3600 B.H.P
11 TREIBER AT 2400 B.H.P.
12 AUGS. SUB. MAX. RATING
13 AUGS. SUB. SU. RATING.
14 AUGS. HIGH SPEED
15 2 CYCLE, D.A
16 FIAT
17 WORTHINGTON, AS RATED
18 SULZER
19 BAND W. D.A. PROPOSED RATING
20 WORTHINGTON AT LIMIT OF PISTON SPEED.
21 NAPIER. W-12.
CURVE 1 DESIGN-"B"
2 DESIGN-"A"
3 ENGINE "C" FIG. 18
4 AUG. SUB. AL. PIST.
5 AUG. SUB.
6 WORTHINGTON.
CURVES OF LIMITING PISTON SPEEDS FOR VARIOUS TYPES OF ENGINES, BASED ON CURVE No. 3 FIG. 5.
5000
4000
3000
2000
1000
0
5
10
15
20
25
30
35
CYL. DIAM. INCHES

FIG. 15

position. This follows from a consideration, not alone of the effect of inertia on bearing loads, but also of the effect of inertia upon the engine framing.

WHAT IS THE LIMIT OF PISTON SPEED?

The conclusion has been reached, from a consideration of the factors of scavenging conditions and engine balance, that piston speeds up to 1800 feet per minute are practicable and that, from the law of similitude, this conclusion holds for all cylinder diameters. The question now arises as to what effect a consideration of the factor of bearing load will have upon limiting the piston speed.

Certain assumptions have been made as a result of a consideration of the data on Figure 5. By making use of these assumptions as to permissible P.V. values for various sized shafts, and remembering that the values for Figures 7, 8, 9 and 13 hold for all cylinder sizes, it is possible to plot curves at limiting piston speeds for various cylinder sizes. These results are shown on Figure 15, where the various curves shown are checked by plotting the results from a number of engines.

These points are for engines designed for a variety of services and have widely different characteristics, but in a general way they tend to show that the human experience curve for piston speeds is not materially different from that for P.V. values on Figure 5.

It follows the general law — Piston Speed $\times$ Cylinder Diameter $=$ Constant, but it must be remembered that the points for the smaller cylinders represent the daring practice of the automotive engineer, and those for the larger engines represent the conservative practice of the marine engineer. Point No. 16, representing the hope of the Augsburg designers, is not far from the results shown by the curves for two-cycle double acting engines.

Taking the least favorable of these curves, No. 3, which is for an engine having a total weight of reciprocating parts 50 per cent greater than the minimum possible, we may assume that from the standpoint of bearing loads a piston speed of 1800 feet per minute is practicable up to the limits covered in Figure 15.

WHAT VALUE OF MEAN INDICATED PRESSURE IS PRACTICABLE FOR HIGH PISTON SPEEDS?

In discussing scavenging conditions, the conclusion was reached that the same degree of scavenging efficiency found in the Augsburg engine could be maintained up to something more than 1800 feet piston speed, by using larger scavenging ports and higher scavenging pressures.

The highest mean indicated pressure developed in that engine was about 95 pounds per square inch. This was in the single cylinder experimental engine from which the present type was developed. For the conditions for which this engine was designed, a higher pressure was not necessary or desirable.

An examination of the results of a number of two-cycle engines justifies the assumption that the mean indicated pressure, based on the effective stroke, is fully as great as that of a four-cycle engine based on the actual stroke. Since the ratio of effective stroke to stroke in this engine is approximately 80 per cent the corresponding mean effective pressure for a four-cycle engine would be about 119 pounds, which is moderate.

The Augsburg submarine engine develops about 130 pounds at maximum power for which the corresponding two-cycle value would be 98 pounds with a ratio of effective stroke to stroke of 75 per cent.

In the trials of the British Admiralty one-cylinder engine, a mean indicated pressure of 150 pounds was developed, which, with a 75 per cent ratio, would give 112 pounds mean indicated pressure for a two-cycle engine.

In view of the fact that for overload conditions some smoking at the exhaust is permissible, a fair assumption for the value of the mean indicated pressure in a two-cycle engine would be 110 pounds per square inch.

The above are for air injection engines. With the development now taking place, and with Sulzer, Augsburg, and a number of others building solid injection engines, there can be no doubt as to the fact that this form of fuel injection has arrived.

Since the injection air is available to support combustion, the value of the mean indicated pressure would, in a solid injection

engine, be reduced to about 87 per cent of the above, which would give about 96 pounds per square inch.

In a four-cycle solid injection engine 119 pounds per square inch has been obtained by Hesselman. With the factor of 75 per cent the two-cycle value would be 89 pounds per square inch.

In the Sulzer experimental double acting engine, with supercharging, a mean indicated pressure of 120 pounds per square inch was developed with air injection, and 103 pounds per square inch was developed with solid injection, the ratio being 86 per cent.

In view of the above a fair assumption for the value of mean indicated pressure would be 90 pounds per square inch, which, with 90 per cent mechanical efficiency possible in a double acting engine with independent auxiliaries, would give a value of 81 pounds per square inch for the mean effective pressure referred to B.H.P.

THE WEIGHT QUESTION.

Dr. Lucke, in a paper read before the Metropolitan Section of the Society of Automotive Engineers, makes the following pertinent comment:

"One of the old doubts heard for many years was concerned with the weight question. Those who had learned that the old marine Diesel engine of the Burmeister and Wain type, for example, weight 450 pounds per H.P., jumped to the conclusion that no Diesel engine could be made light enough for automotive service. For a long time no analysis was made of the prime factors, and the shock of learning that the large M.A.N. submarine engines weighed 50 pounds per H.P. was survived by the still vague conclusion that this must surely be the low limit. This was all nonsense, of course, and more people are now learning the elements of weight well enough to accept the statement that an automotive Diesel engine can be made to weigh not more than 30 per cent more than an automotive gasoline engine, though not many are ready to believe that there is no inherent reason why it may not be made actually lighter than the gasoline engine without reducing the engine life, even if no one has so far succeeded in doing this."

The weight per B.H.P. may be expressed by the simple basic formula:

$$\text{Pounds per B.H.P.} = \frac{\text{pounds cubic inch} \times \text{stroke inches} \times 33{,}000}{\text{m.e.p.} \times \text{piston speed}}$$

The uncertain factor in the above equation is the value of weight per cubic inch cylinder volume. It has been shown by the data of Figure 1 that, assuming a basic value of 4.5 pounds for a 20-inch cylinder, there should be no material change in that value, even assuming the doubtful assumption that foundry limitations require larger casting sections for small engines than strength considerations demand.

The above figure was arrived at from an examination of the published weights of the high speed, two-cycle double acting Augsburg engine.

The weights of this engine are given at 4.1 pounds per cubic inch cylinder volume, but this evidently does not include the weight of the independently driven scavenging blower. A fair allowance for this weight would be 10 per cent of the engine weight, which would bring the total up to the figure used.

This engine is a cast-iron engine built for high speed and is presumably sufficiently rigid for the service intended.

In view, however, of the materially higher piston speeds proposed, the matter of rigidity can not be overdone. It is believed, however, that by the use of steel castings, and with no attempt to reduce the weight thereby, this figure may be taken as sufficient.

CONCLUSIONS.

The following conclusions may be drawn and stated as a basis for determining the results which may be expected:

1. That, owing to the very material increase in the port area possible the scavenging system used by Tartrais offers superior advantages over any other system in use.

2. That, with a possible scavenging pressure of 3 pounds per square inch, the scavenging efficiency necessary to the development of 90 pounds per square inch mean indicated pressure can be

maintained up to something more than 1800 feet per minute piston speed.

3. That, the P. V. values shown on curve 3, Figure 5, are conservative and that a consideration of the bearing load permits piston speeds higher than those at which the scavenging efficiency can be maintained.

4. That a weight per cubic inch cylinder volume of 4.5 pounds is ample to provide the rigidity necessary for the development of the high piston speed required, and that between the limits of a 12-inch cylinder diameter and a 30-inch cylinder diameter there will be no variation in this value due to foundry limitations.

RESULTS WHICH MIGHT BE EXPECTED FROM A COMBINATION OF THE FACTORS CONSIDERED.

The characteristics of a two-cycle double acting engine with a stroke bore ratio of 1.5 are shown on Figures 16 and 17.

For the purpose of comparison the characteristics of a four-cycle single acting trunk piston engine are shown on Figure 16.

The curves of P.V. values are those for Design "B" from Figure 9, and for the Augsburg submarine engine, with light reciprocating parts, from curve 2, Figure 6. These two curves, representing the results for engines with the lightest possible reciprocating weights, have been extended beyond the limits of Figures 6 and 9.

This has been done with a fair amount of accuracy by using the data of curves for B.H.P. per cylinder.

Since these curves start from zero they can be drawn with fair accuracy for the small cylinder diameters with large P.V. values.

A comparison of P.V. curves shows very clearly the marked advantage of the two-cycle double acting engine up to about 1800 feet per minute piston speed. Above that point, due to the heavier reciprocating parts, the rise in the P.V. values becomes abnormal.

The curves for limiting piston speeds are based on the P.V. values for the crank pin load. Consideration of this factor alone would permit piston speeds higher than 1800 feet per minute for all cylinder sizes. Since, however, caution applied to other factors limits the piston speed, the curves of B.H.P. per cylinder and the

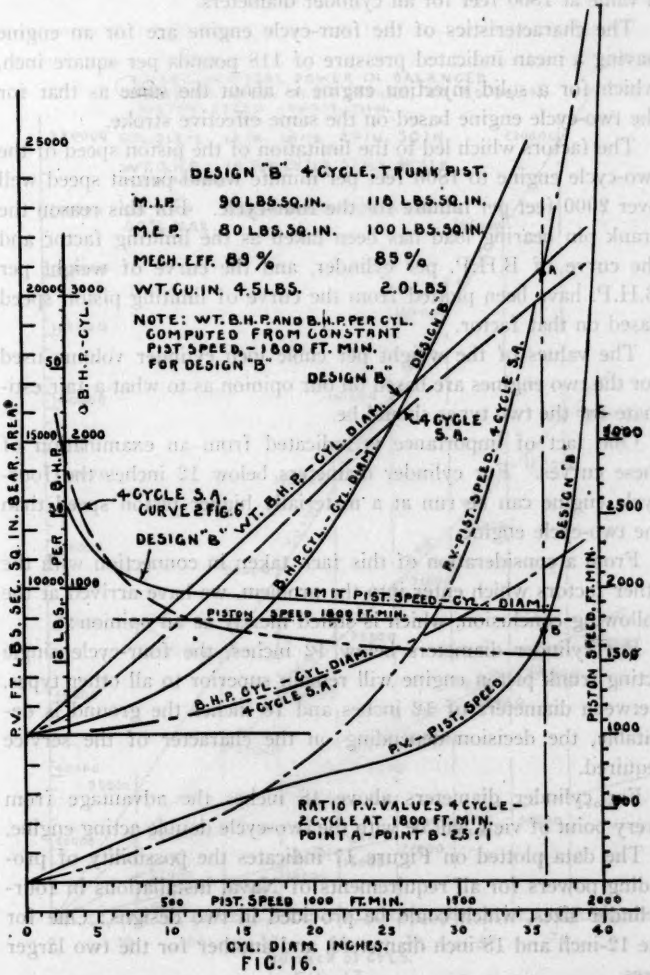

DESIGN "B" 4 CYCLE, TRUNK PIST.
M.I.P. 90 LBS.SQ.IN. 118 LBS.SQ.IN.
M.E.P. 80 LBS.SQ.IN. 100 LBS.SQ.IN.
MECH. EFF. 89% 85%
WT. CU. IN. 4.5 LBS. 2.0 LBS
NOTE: WT. B.H.P. AND B.H.P. PER CYL. COMPUTED FROM CONSTANT PIST SPEED, - 1800 FT. MIN. FOR DESIGN "B"
DESIGN "B"
4 CYCLE S.A.
4 CYCLE S.A. CURVE 2 FIG. 6
DESIGN "B" WT. B.H.P. - CYL. DIAM.
B.H.P. CYL. - CYL. DIAM.
P.V. - PIST. SPEED. 4 CYCLE S.A.
DESIGN "B"
LIMIT PIST. SPEED - CYL. - DIAM.
PISTON SPEED 1800 FT. MIN.
B.H.P. CYL. - CYL. DIAM. 4 CYCLE S.A.
P.V. - PIST. SPEED
RATIO P.V. VALUES 4 CYCLE - 2 CYCLE AT 1800 FT. MIN. POINT A - POINT B - 2.5 : 1
P.V. FT. LBS. SEC. SQ. IN. BEAR. AREA
B.H.P. PER CYL.
PISTON SPEED FT. MIN.
PIST. SPEED FT. MIN.
CYL. DIAM. INCHES.

FIG. 16.

weight per B.H.P. for the two-cycle engines have been plotted for a value at 1800 feet for all cylinder diameters.

The characteristics of the four-cycle engine are for an engine having a mean indicated pressure of 118 pounds per square inch, which for a solid injection engine is about the same as that for the two-cycle engine based on the same effective stroke.

The factors which led to the limitation of the piston speed of the two-cycle engine to 1800 feet per minute would permit speed well over 2000 feet per minute for the four-cycle. For this reason the crank pin bearing load has been taken as the limiting factor and the curve of B.H.P. per cylinder, and the curve of weight per B.H.P. have been plotted from the curve of limiting piston speed based on that factor.

The values of the weight per cubic inch cylinder volume used for the two engines are based on our opinion as to what a fair estimate for the two types should be.

One fact of importance is indicated from an examination of these curves. For cylinder diameters below 12 inches the four-cycle engine can be run at a materially higher piston speed than the two-cycle engine.

From a consideration of this fact, taken in connection with the other factors which enter into the problem, we have arrived at the following conclusion, which is stated merely as an opinion:

For cylinder diameters below 12 inches, the four-cycle single acting trunk piston engine will remain superior to all other types. Between diameters of 12 inches and 18 inches the ground is debatable, the decision depending on the character of the service required.

For cylinder diameters above 18 inches the advantage from every point of view will be with the two-cycle double acting engine.

The data plotted on Figure 17 indicates the possibility of providing powers for all requirements of Naval installations in four-cylinder sizes, which could be provided in two designs: One for the 12-inch and 18-inch diameters, and another for the two larger sizes.

The foregoing discussion has resolved itself into a study of piston speed, which is natural in view of the fact that this is the

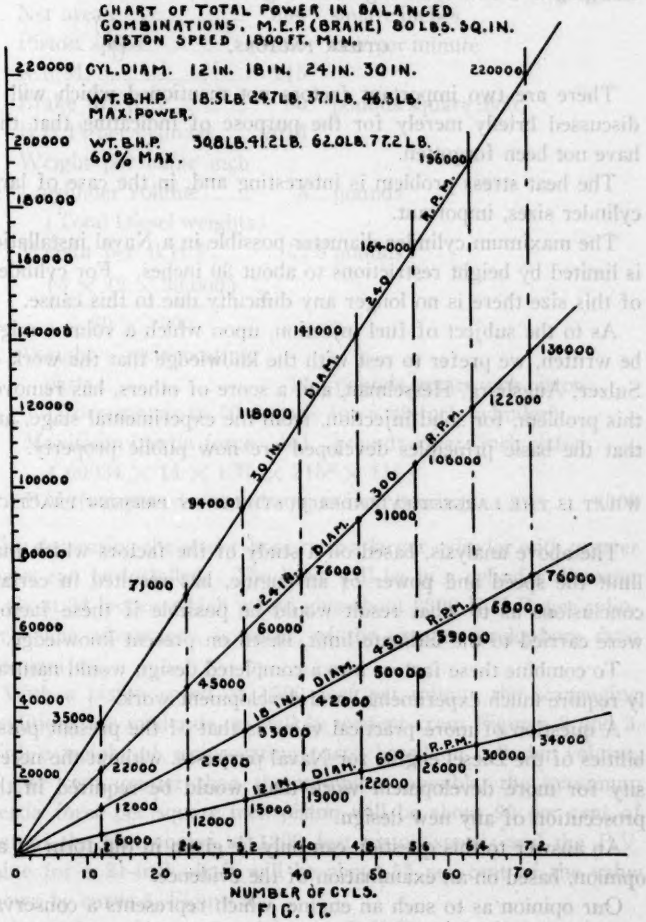


CHART OF TOTAL POWER IN BALANCED COMBINATIONS. M.E.P.(BRAKE) 80 LBS. SQ.IN. PISTON SPEED 1800 FT. MIN.
CYL. DIAM. 12 IN. 18 IN. 24 IN. 30 IN.
WT. B.H.P. MAX. POWER. 18.5 LB. 24.7 LB. 37.1 LB. 46.3 LB.
WT. B.H.P. 60% MAX. 30.8 LB. 41.2 LB. 62.0 LB. 77.2 LB.
220000
200000
180000
160000
140000
120000
100000
80000
60000
40000
20000
0
10
20
30
40
50
60
70
NUMBER OF CYLS.
R.P.M.
DIAM.
196000
164000
141000
118000
94000
71000
60000
45000
35000
25000
22000
19000
15000
12000
6000
136000
122000
106000
91000
76000
68000
59000
50000
42000
33000
30000
26000
22000
15000
12000
6000
34000
30000
26000
22000
19000
15000
12000
6000
2.4
3.2
4.0
4.8
5.6
6.4
7.2
30 IN.
24 IN.
18 IN.
12 IN.

FIG. 17.

one factor in which a material increase is justified by present knowledge. The mean indicated pressure used is the highest which is considered practicable without supercharging, and supercharging in a two-cycle engine belongs to the future.

OTHER FACTORS.

There are two important factors not mentioned which will be discussed briefly merely for the purpose of indicating that they have not been forgotten.

The heat stress problem is interesting and, in the case of large cylinder sizes, important.

The maximum cylinder diameter possible in a Naval installation is limited by height restrictions to about 30 inches. For cylinders of this size there is no longer any difficulty due to this cause.

As to the subject of fuel injection, upon which a volume might be written, we prefer to rest with the knowledge that the work of Sulzer, Augsburg, Hesselman, and a score of others, has removed this problem, for solid injection, from the experimental stage, and that the basic principles developed are now public property.

WHAT IS THE LARGEST CYLINDER JUSTIFIED BY PRESENT PRACTICE.

The above analysis, based on a study of the factors which may limit the speed and power of an engine, has resulted in certain conclusions as to what result would be possible if these factors were carried to the ultimate limit, based on present knowledge.

To combine these factors into a completed design would naturally require much experimental and development work.

A question of more practical value is that of the present possibilities of the Diesel engine for Naval purposes, without the necessity for more development work than would be required in the prosecution of any new design.

An answer to this question can only be given in the form of an opinion, based on an examination of the evidence.

Our opinion as to such an engine, which represents a conservative estimate as to the largest cylinder which can be installed within the height limitation of a capital ship, is as follows:

Stroke	42	inches = 3.5 feet
Ratio $\frac{\text{stroke}}{\text{bore}}$	1.5	
Cylinder diameter	28	inches
Net area	554	square inches
Piston speed	1505	feet per minute
R.P.M.	215	
Brake m.e.p.	80	pounds square inch
B.H.P. per cylinder	2015	
Weight per cubic inch cylinder volume (Total Diesel weights)	5	pounds
Weight per B.H.P. $\frac{(5 \times 42 \times 33{,}000)}{80 \times 1500}$	57.5	pounds
Weight reciprocating parts	14	pounds square inch piston (Corresponds to 10 pounds for a 20-inch cylinder)
Maximum inertia force $(.00034 \times 14 \times 1.75 \times 215^2 \times 1\frac{1}{4})$	481	pounds square inch piston
P.V. foot pounds second square inch bearings	6000	

By decreasing the stroke-bore ratio a larger cylinder with greater power can be installed. The limit will be in a cylinder diameter of about 34 inches, which would give about 2900 B.H.P. per cylinder. The advantages, however, of the greater stroke-bore ratio are manifest.

With a piston speed of 1500 feet per minute the scavenging conditions are moderate, as will be evident from Figures 2 and 3. With a weight of reciprocating parts, based on cylinder volume, 25 per cent greater than the minimum possible, the maximum inertia force per square inch piston will be about 90 per cent of that of the Fiat engine at 1200 feet piston speed; and the P.V value for a 21-inch shaft will be about 55 per cent of the value shown by curve 3, Figure 5.

As to the estimate of weight per B.H.P. we invite attention to the pertinent remarks of Dr. Lucke, previously quoted.

The above engine rated on the basis of 60 per cent of its full power rating would have the following characteristics:

R.P.M. (assuming that power varies as cube of R.P.M.)	181
Piston speed	1267 feet per minute
B.H.P. (60 per cent × 2000)	1200
M.e.p.	56 pounds square inch
M.I.P. at 85 per cent mechanical efficiency	67 pounds square inch
Weight per B.H.P.	96 pounds

U. S. COAST GUARD CUTTERS, *CHELAN, PONTCHARTRAIN, TAHOE, CHAMPLAIN,* AND *MENDOTA.*

By Commander Q. B. Newman, U. S. Coast Guard, Member.

This Journal in its issue of November, 1921, carried a general description of a new type of propelling machinery, together with the reasons which had led to its adoption for Coast Guard vessels.

The four Coast Guard Cutters of the *Tampa* class, built in 1919 — 1921, were pioneers, in that they were the first ships for which turbo-electric propulsion with synchronous motor drive was adopted. In the seven years intervening, the performance of the new type of machinery has amply vindicated itself under the rigorous conditions of Coast Guard operation.

The *Tampa* class was closely followed by the *Cuba,* a small passenger ship; the *San Benito,* a British-built United Fruit Company ship; and the *Kamoi,* a Japanese fuel ship. Then came a lull in synchronous motor installations, which was broken by the startling announcement that the Panama-Pacific liner *California,* the largest commercial vessel ever built in the United States, would be propelled by two synchronous motors of 8500 shaft horsepower each. The *California* has been in commission since the first of this year. The same line is building two additional ships, somewhat larger than the *California,*— the *Virginia,* which was launched in August, and a third [name not yet assigned] which has been laid down; and they also have synchronous motor propulsion.

A large liner now building in England is to have American-built turbo-electric drive with synchronous motors.

Without wishing to appear as a propagandist for or against any type of propelling machinery, the writer believes that for Coast Guard vessels no other present type is superior to turbo-electric synchronous motor drive when considering a combination of reliability, fuel economy, first cost, weight, space, maintenance cost, and continuous availability for service, as these factors are related

for Coast Guard cutters. There was, therefore, no hesitancy in selecting that type of propulsion for the five cutters now building at the Fore River plant of the Bethlehem Shipbuilding Corporation. For other types or sizes of ships with other types of duty, the factors mentioned might be weighted differently and other factors might enter; so that this form of drive is not the universal answer to the question of type of propulsion.

But the writer is willing to appear as a propagandist against the waste of steam in the conventional types of steam-driven auxiliaries commonly installed. By comparison with this waste, the difference between types of propelling machinery is hardly worth discussing. This statement is based on complete trials on the *Modoc*, one of the *Tampa* class, which are described in Transactions Society of Naval Architects and Marine Engineers, Vol. 31, 1923. On the *Modoc* every pound of steam was traced to its ultimate destination, and it was found that at full speed the auxiliaries, including excitation of main machinery, consumed 32 per cent of all the steam produced; and at usual cruising speed, 42½ per cent. Of course, some of the exhaust steam was used in the feed heater and some found its way to the lower stages of the main turbine, but even so, the performance as a whole was not flattering.

All of which is recited here to lead up to the statement that no ship can be called efficient if a large part of her machinery is extravagant; and as a prelude to the question: — How can real economies be effected in driving auxiliary machinery?

During the three years following the *Modoc* trials, no steam vessels were built for the Coast Guard, and the engineering division of the Service embraced the opportunity to work out at least one answer: — Do it as it is done on shore. The owner of an office building, or a residence, or a factory, does not provide a separate steam machine to furnish power for each of many requirements; but he runs two or three wires into his place and takes his power from a central power station. We have a central power station on any electrically propelled ship,— why not use it? There was, of course, the usual number of reasons why not, and more than the usual number of things that could not be done. Which makes it necessary to anticipate the final answer and announce that it could be done. The *Chelan* and the *Pontchartrain* have had their

trials, and before this article is printed the *Tahoe* will have been completed and put in commission.

The propelling machinery gave little concern because, as already stated, the type had been abundantly demonstrated. Also it was known that auxiliary machinery could be motor driven, because motorship machinery had forced the marine engineering fraternity to ascertain that fact. The real problem was to tie up machinery whose speed ought to be constant, to the main alternating current generator whose speed must necessarily be varied to vary the speed of the ship; to obtain constant voltage for ship's lighting; and to obtain direct current for excitation, battery charging, radio, and other purposes.

It was the direct current need that suggested the solution. That, obviously, called for a motor-generator; the motor being an A.C. motor taking current from the main generator, and therefore working through a wide range of speeds; the D.C. generator being driven through the same range of speeds, but giving a constant voltage output. Two systems had already been worked out to accomplish the latter result, and both had been in use on shore for several years. The general principle is the same in both. The generator is designed for full power and voltage at its lowest speed. For higher speeds, the excitation is automatically reduced as necessary for constant voltage. On these ships the working range of the D.C. generator is from 800 to 1200 R.P.M., which corresponds to 40 to 60 cycles of the main generator.

But the main generator will frequently run below 40 cycles, and in port it will not run at all. A geared turbine is therefore connected to the shaft of the motor-generator, for use when main generator speed is below 40 cycles. The auxiliary turbine rotates idly in vacuo when the main generator is furnishing the power, and the windage loss is just about sufficient to keep the turbine casing hot and in condition for full influx of steam when the cross connection to the main set is broken. Because of the necessity of getting main and auxiliary generators synchronized and in phase before inter-connecting, the connection is made manually. The disconnecting is accomplished automatically through the medium of the governor oil pressure of the auxiliary turbine. This pressure is maintained by a centrifugal pump on the pinion shaft. At

60 cycles the oil pressure is 75 pounds per square inch, and at 40 cycles, 36 pounds. The latter pressure is insufficient to hold the governor valve mechanism closed, so the valve admits steam to the auxiliary turbine; and, after admitting steam, trips a circuit breaker in the inter-connection. Main and auxiliary sources of power are then independent until the operator again closes the inter-connecting switch.

A vertical section of the governor mechanism is shown in Figure 1. The flange at the bottom of the drawing is bolted to the body of an ordinary globe valve, and the stem 4 is extended down and attached to the valve disc. The governor valve is opened or closed by the rise or fall of the piston 1, to which valve stem 4 is attached. Piston 1 is floating between the compression of spring 3 and the pressure of oil entering the cylinder at A, so that if the oil pressure is below normal, *i. e.*, if the turbine is running below normal speed, spring 3 raises the piston, thereby opening the governor valve wider for more steam; if the oil pressure is above normal, *i. e.*, if the turbine is running above normal speed, oil pressure acting on the piston overcomes spring 3 and chokes down the governor valve. Compression on the spring can be varied at will by raising or lowering spring seat 8 by means of screw 7.

The governor valve can be closed entirely by admitting oil to cylinder F where it can act on the step 2 of piston 1. This occurs when the auxiliary load is thrown on the main generator, and is brought about in the following manner: — Suppose the main generator is running at 50 cycles, and the auxiliary generator at 60 cycles. The first step is to reduce the auxiliary generator speed to 50 cycles, which is done by lowering spring seat 8. Governor oil pressure entering at A holds piston 10 up against compression of spring 11. There is a continuous slow leakage of oil past piston 10 into port D, which establishes a pressure in cylinder H sufficient to hold piston 14 at the top of its stroke and cause piston valve 16 to assume the position shown. In this position no oil can enter cylinder F. When the frequency of the auxiliary generator has been made the same as the frequency of the main generator, and the two systems are in phase, the inter-connecting switch is closed and the two systems are in parallel. But in the act of closing the

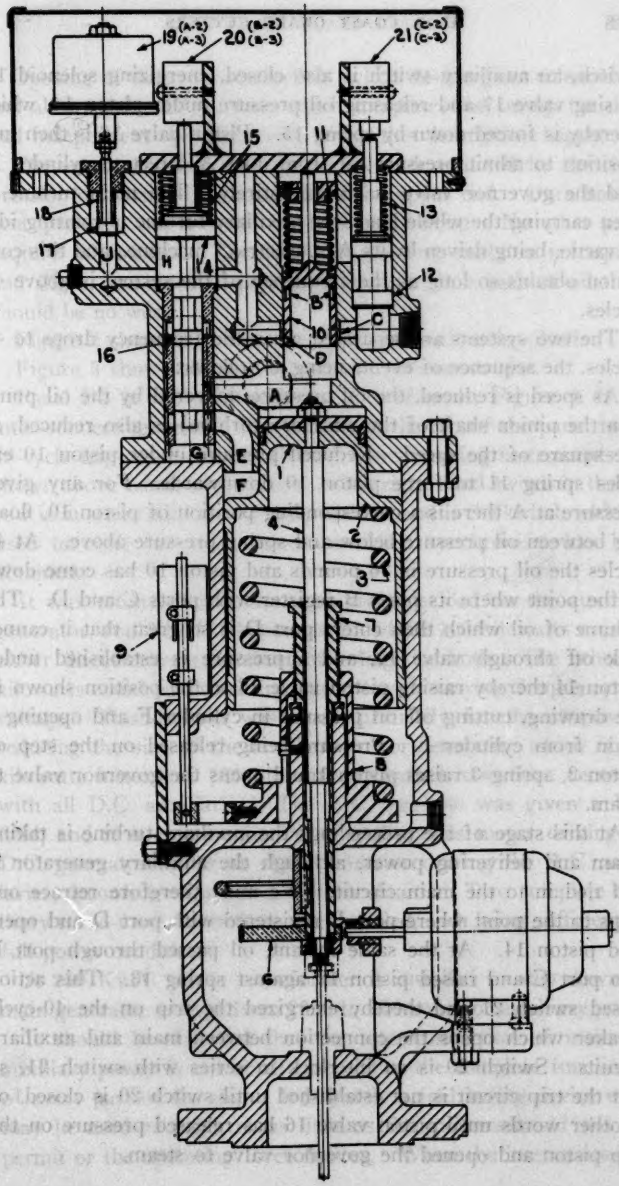


FIG. 1.—GOVERNOR AND CONTROL MECHANISM, AUXILIARY TURBINE.

switch, an auxiliary switch is also closed, energizing solenoid 19, raising valve 17 and releasing oil pressure under piston 14, which thereby is forced down by spring 15. Piston valve 16 is then in a position to admit pressure oil from inlet A direct to cylinder F, and the governor valve is closed entirely. The main turbine is then carrying the whole load; the auxiliary turbine is rotating idly in vacuo, being driven by its A.C. electrical machine; and this condition obtains so long as the frequency of the system is above 40 cycles.

The two systems are separated when the frequency drops to 40 cycles, the sequence of events being as follows:

As speed is reduced, the oil pressure delivered by the oil pump (on the pinion shaft of the auxiliary turbine) is also reduced, as the square of the speed. Reduced pressure under piston 10 enables spring 11 to force piston 10 downwards. For any given pressure at A there is a corresponding position of piston 10, floating between oil pressure below and spring pressure above. At 40 cycles the oil pressure is 36 pounds and piston 10 has come down to the point where its ports B register with ports C and D. The volume of oil which then enters port D is so great that it cannot leak off through valve 17, and a pressure is established under piston 14 thereby raising piston valve 16 to the position shown in the drawing, cutting off oil pressure in cylinder F and opening a drain from cylinder F. Pressure being released on the step of piston 2, spring 3 raises piston 1 and opens the governor valve to steam.

At this stage of the proceedings the auxiliary turbine is taking steam and delivering power, although the auxiliary generator is still tied in to the main circuit. We must therefore retrace our steps to the point where port B registered with port D and operated piston 14. At the same instant, oil passed through port B into port C and raised piston 12 against spring 13. This action closed switch 21 and thereby energized the trip on the 40-cycle breaker which opens the connection between main and auxiliary circuits. Switch 20 is an interlock in series with switch 21, so that the trip circuit is not established until switch 20 is closed, or in other words until piston valve 16 has released pressure on the step piston and opened the governor valve to steam.

Opening of the cross-connecting switch de-energizes solenoid 19 and allows valve 17 to close, which completes the set-up for normal independent operation with which this description began.

The speed at which trip-out occurs is determined by the compression on spring 11, adjustable at the upper end. Once having been properly set, it needs no further attention. No other parts require adjustment. The travel of moving parts is infrequent and slight, and since all parts are immersed in a bath of oil there should be no wear.

Figure 2 shows all of the electrical connections involved.

Figure 3 shows one of the auxiliary generators sets.

Panel I-2 in the photograph of switchboards, Figure 4, is the inter-connecting panel. The breaker at the top of the panel is the 40-cycle trip-out above described. Synchronizing instruments are shown above center of the panel. The right-hand switch in the top row is in series with the 40-cycle breaker and is the switch which is used to close the inter-connection when synchronism has been obtained.

It was estimated that the total probable load on the auxiliary generator would be 185 kilowatts, of which 100 kilowatts must be either direct current or constant voltage or both. The other 85 kilowatts could be alternating current with varying frequency and voltage. This division was made, primarily for the purpose of reducing the size of the auxiliary motor-generator, which is 100 kilowatts instead of 185 kilowatts as would have been necessary with all D.C. auxiliaries. The A.C. machine was given a D.C. field, and it operates either as a synchronous motor driving the D.C. generator, or as an A.C. generator driving the A.C. auxiliaries, according to which condition obtains. Two complete auxiliary turbo-motor-generators are installed, each being sufficient for full power conditions.

A 250 Kv.A. power transformer is installed to step down the main generator voltage from 2300 to 230 volts for auxiliary power.

It should be observed that the inter-connection of main and auxiliary power is not at all essential to the complete functioning of every part of the ship. It is purely a fuel saving device which may be used as much or as little as the movements of the ship permit or the operating force desires. When the inter-connecting

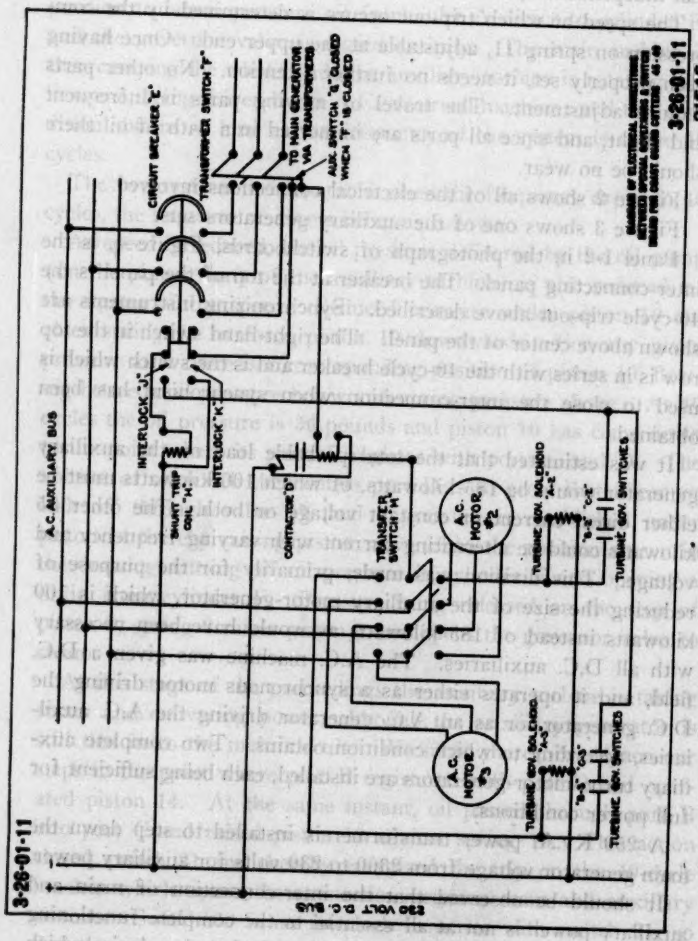


FIG. 2.—INTERCONNECTION BETWEEN MAIN GENERATOR AND AUXILIARY GENERATOR.

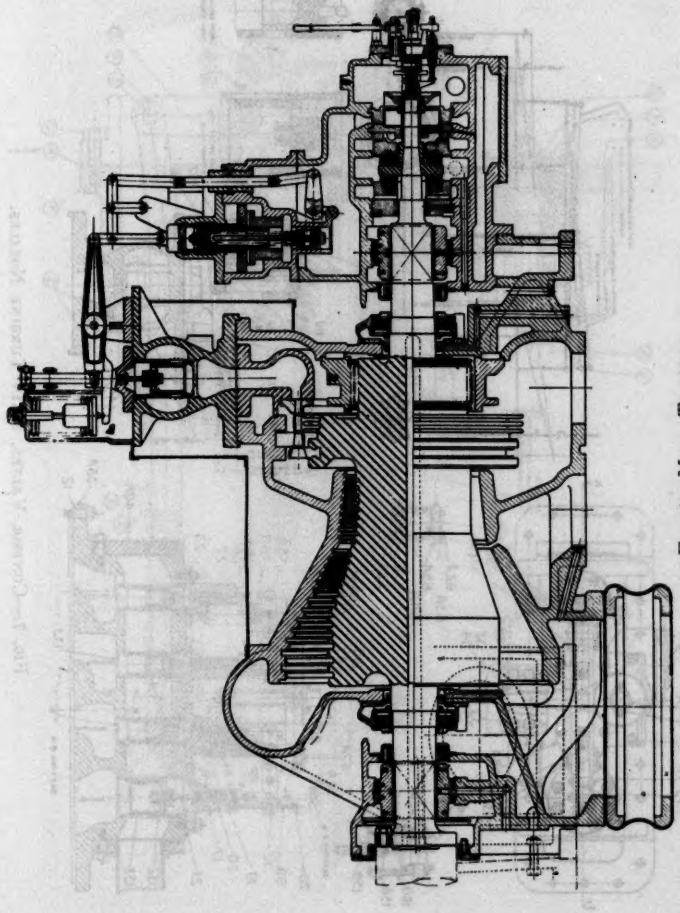


FIG. 6.—MAIN TURBINE.

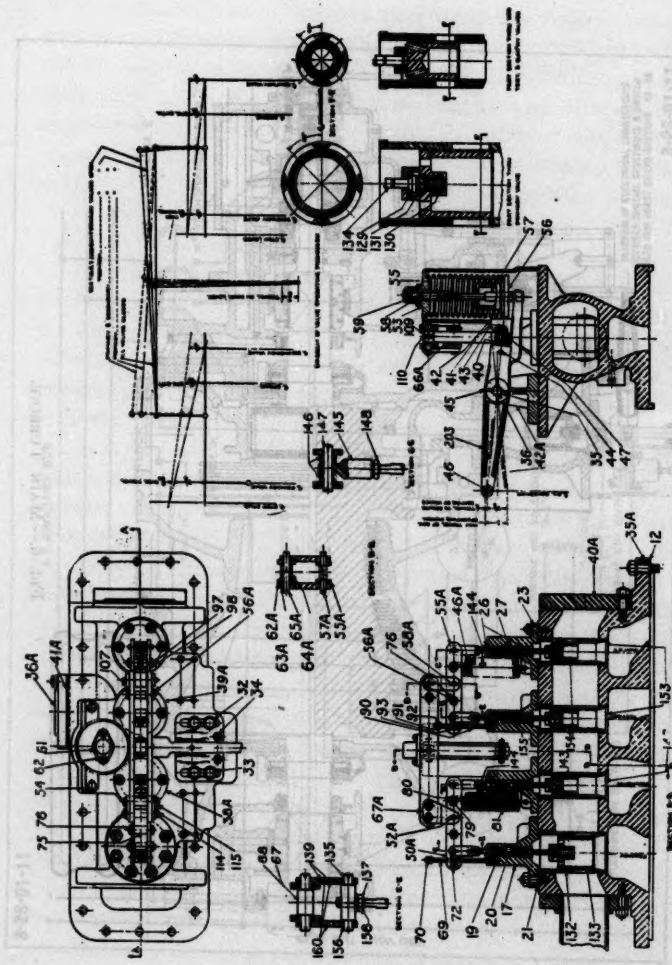


FIG. 7.—CONTROL VALVES, MAIN TURBINE NOZZLES.

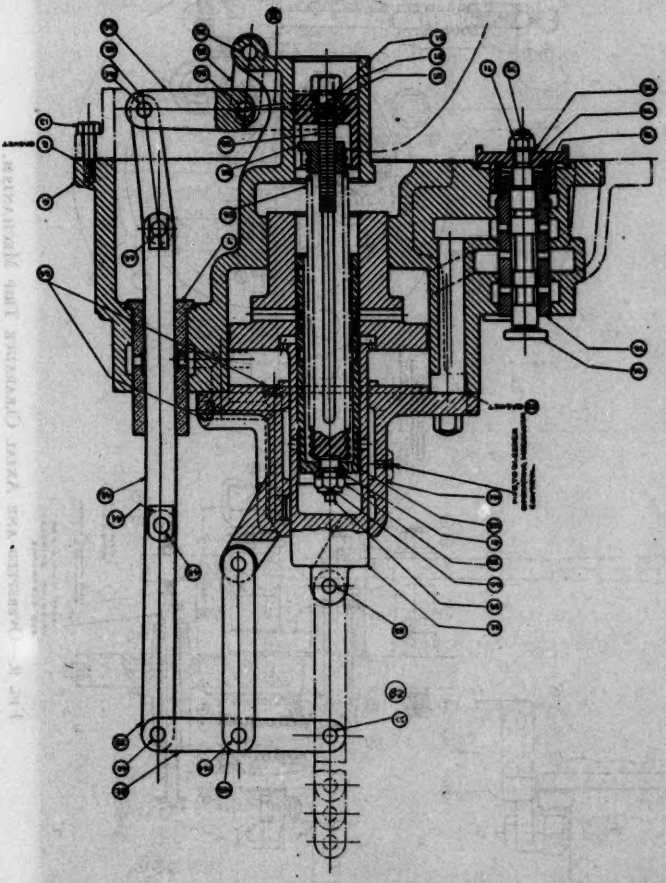


FIG. 8.—HYDRAULIC GOVERNOR, MAIN TURBINE.

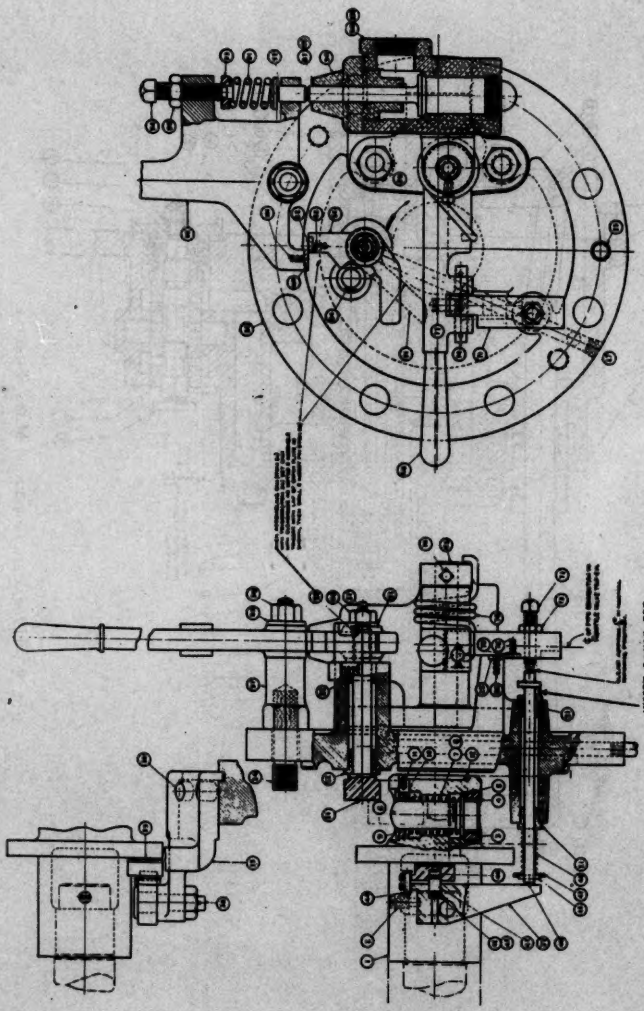


FIG. 9.—OVERSPEED AND AXIAL CLEARANCE TRIP MECHANISM.

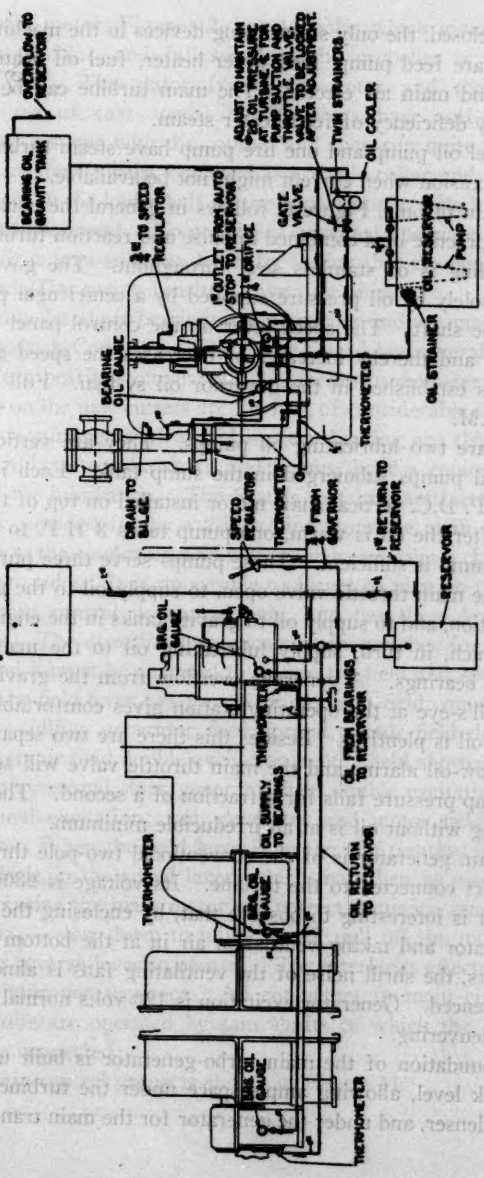


FIG. 10.—LUBRICATING SYSTEM—MAIN TURBO—GENERATOR.

switch is closed, the only steam using devices in the machinery department are feed pump, feed water heater, fuel oil heater, main turbine, and main air ejector. The main turbine can be bled to supply any deficiency of feed heater steam.

One fuel oil pump and one fire pump have steam turbine drive for any occasion when current might not be available.

The main turbine, Figure 6, follows in general the usual Westinghouse practice of a combined impulse and reaction turbine. The rotor blading is of stainless steel throughout. The governor is actuated solely by oil pressure supplied by a centrifugal pump on the turbine shaft. The speed lever on the control panel operates a leak-off and thereby increases or decreases the speed at which stability is established in the governor oil system. Full speed is 3600 R.P.M.

There are two lubricating oil pumps. They are vertical shaft centrifugal pumps submerged in the sump tank. Each is driven by a 5 H.P. D.C. vertical shaft motor installed on top of the sump tank. After the oil is warm, one pump takes 3 H.P. to drive it, and one pump is sufficient. These pumps serve three purposes,—to hold the main throttle valve open, to supply oil to the governor pump suction, and to supply oil to gravity tanks in the engine-room trunk, which, in turn, supply lubricating oil to the main turbo-generator bearings. A generous overflow from the gravity tanks past a bull's-eye at the operating station gives comfortable assurance that oil is plentiful. Besides this there are two separate systems of low-oil alarm; and the main throttle valve will snap shut if the pump pressure fails for a fraction of a second. The danger of running without oil is at an irreducible minimum.

The main generator is of the conventional two-pole three-phase type, direct connected to the turbine. Its voltage is 2300 at full speed. It is interesting to observe that, by enclosing the ends of the generator and taking ventilation air in at the bottom through short ducts, the shrill noise of the ventilating fans is almost completely silenced. Generator excitation is 120 volts normal and 240 volts maneuvering.

The foundation of the main turbo-generator is built up to the berth deck level, allowing ample space under the turbine for the main condenser, and under the generator for the main transformer.

The main motor, Figure 12, has 44 poles which gives a speed reduction of 22:1, and full speed of the propeller is therefore 163.5 R.P.M. The stator frame, the rotor and the bearing brackets are of cast steel. The bearings have self-contained lubricating systems with disc oiling, and they run quite cold.

The Kingsbury thrust bearing has a single collar and there are thrust shoes in the lower half of the housing only. Its lubrication also is self-contained. No cooling is provided and none is necessary, for it is impossible to detect by feeling that the bearing is in operation. The spring bearings have ring oiling.

The propeller shaft bearings are of rubber and are made by the B. F. Goodrich Company. The Coast Guard has several hundred rubber stern bearings on shafting 3½ inches in diameter and less, but those on the new cutters are the first of considerable size. The essential requirements of a rubber stern bearing are that it must not be a tight fit on the shaft, and it must have a copious supply of water for lubrication. The first is met in manufacture. The second is provided for by a 2-inch pipe from the main circulator discharge to the condenser, and with no valves except a check valve at the stern tube. It is impossible to forget to turn on the water.

The main control is manual only, and has three levers with interlocks. The direction lever closes main switches for ahead or astern, and it must be operated to release the field and speed interlocks. The field lever has four positions with field connections as follows: — idling, generator field normal, motor field dead; first start, generator field double excitation, motor field shorted through a resistance; second start, generator field double excitation, motor field normal excitation; run, generator and motor fields normal excitation. When the field lever is in the run position it releases the interlock on the speed lever, which may then be moved. To stop or reverse the main motor the correct sequence must be followed, viz.,— slow down to idling speed, pull off the motor field, and bring reverse lever to neutral. The interlocks effectually prevent an improper sequence. All contactors in main circuit and field circuits are operated by cam shafts to which the operating levers are connected.

The main control group is shown in Figure 11.

As between A.C. and D.C. motors for auxiliary power, the most that can be said for D.C. is that it is an excellent machine and that it is measurably well known. Its chief disadvantage is due to the fact that the heavy current is carried by the armature, which is the rotating part, rendering it necessary to pass the heavy current through a sliding contact between brushes and commutator. The current actually passing through the windings of a D.C. machine is in reality an alternating current, for which the commutator acts as a rectifier. Flat or loose commutator bars, protruding mica strips, or poorly fitted brushes produce sparking, which starts a new line of trouble. A load which is heavy enough to stall the motor will generally cause the commutator to flash from brush to brush. And yet with reasonable care and attention the D.C. motor will give excellent service.

But the A.C. motor is a much better machine, particularly for marine service (where its employment is most rare), for the reason that it will come through an accidental shower bath without injury. In this machine the heavy current is carried by the windings of the stationary part, and, therefore, does not pass through sliding contacts.

There are two general types of A.C. motors,—synchronous and induction. The synchronous motor has a direct current field and must therefore be separately excited. The field poles are bolted to the outer surface of the rotating part, and the exciting current, which is 1½ to 2 per cent of the power, is supplied through brushes which bear on slip rings on the shaft. Item for item a synchronous motor is a direct current motor turned inside out, except that the slip rings are not segmental like a commutator.

On the new cutters, synchronous motors are used for propulsion, and for the auxiliary turbo-motor-generator sets. They have extremely good efficiency, and are of light weight and relatively small size.

While on the subject of synchronous motors it should be remarked that a very real advantage lies in their accessibility for repairs. The propelling motor, for example, could be entirely rewound without disassembling, and without lifting anything heavier than a field pole, which weighs 200 pounds. On the *Tampa* class during seven years service of four ships, the only

repairs to propelling motors has consisted of renewing one pair of poles on one ship.

The other general type of A.C. motor is the induction motor, which has its field induced in the rotating part by the action of the current in the coils of the stationary armature. The rotor windings may consist of naked copper bars imbedded in the iron and connected to two copper end rings. This winding accounts for the name of the squirrel cage motor. The rotor bars are not insulated from the iron and are short circuited through the end rings. The squirrel cage motor is surpassed in ruggedness only by an anvil, and the only attention it requires is in the matter of bearing lubrication. Its two drawbacks are that its starting torque is poor and its speed cannot be regulated.

In the other form of induction motor the rotor windings are applied in much the same way as the armature windings of a D.C. machine. The ends are brought out to slip rings on the shaft. From the slip rings secondary current passes through brushes and a controller to a bank of resistances. With properly proportioned resistances the starting torque can be made to serve any reasonable need, and speed can be regulated as desired. The starting, reversing and speed control devices for a wound rotor induction motor are very much simpler than the corresponding devices for a D.C. motor. Wound rotor motors are used for windlass and capstan motors on the new cutters. When there is a tendency for the load to overhaul, as in paying out an anchor in deep water, pay-out power is applied to the motor, which prevents the load taking charge.

Brakes for windlass and capstan motors presented an unusual problem. They are mounted in the weather and therefore must be water tight, but an A.C. brake cannot be made watertight. The brake is supposed to apply itself when power is cut off from the motor, so a separate D.C. brake circuit operated by the motor controller would introduce an element of uncertainty. The problem was solved by installing a disc rectifier such as is used for charging radio batteries, and this produces the necessary direct current for actuating the brake.

The anchor windlass has a 30 H.P. motor and it heaves in two 4000-pound anchors with 45 fathoms of 1¾-inch chain each at a

rate of 5.6 fathoms per minute, the power input being 30 kilowatts. It has ten steps of speed from 50 per cent to 100 per cent of rated full speed.

The capstan motor is interchangeable as a whole or part for part with the windlass motor.

Squirrel cage motors are used to drive the forced draft blower, quarters ventilation fans, motor ventilation fan, machine shop, and the following pumps: — main circulating, auxiliary circulating, general service, evaporator feed, and one fuel oil.

The main circulator, the fuel oil pump, and the motor ventilation fan are arranged to be driven by the main generator throughout the entire speed range if desired, on the assumption that their output will be at least adequate to actual needs even when the main generator is slowed to 15 cycles. Two separate savings are thus effected,— their power at low speeds is much reduced, and the cost in steam per horsepower-hour is based on main turbine performance. However, the switching arrangement allows them to be shifted to the auxiliary generator when desired.

Direct current motors are used to drive the refrigerating machines, one fire pump, sanitary pump, main and auxiliary condensate pumps, fresh water pump, lubricating oil pumps, lubricating oil purifier, laundry machinery, and motor-generators for radio, gyro-compass and fathometer.

Ship's lighting and searchlights are direct current.

All A.C. auxiliary power distribution is 3-phase, 230 volts; and nearly all D.C. power distribution is 230 volts.

The main circulating pump is a Warren double inlet centrifugal pump with a capacity of 4000 gallons per minute with a total dynamic head of 15.5 feet. This is one of the auxiliaries which are to be driven by the main generator at all speeds. It will therefore come down to one-quarter speed during maneuvering and standing by. As originally projected, the main overboard discharge was to be 18 inches above the water line. The discharge pressure of a centrifugal pump varies as the square of the speed, and the power as the cube of the speed. It followed that a centrifugal pump which would deliver any water at all at quarter speed through an outlet 18 inches above the water line, would require 39 horsepower when delivering 4000 gallons at full speed. Deep

submergence of the overboard discharge pipe brought the maximum power down to 19.6 horsepower, and a 20-horsepower squirrel cage motor is used to drive the pump.

The bilge pumping system is one which has come through a gradual evolution in the Coast Guard. The apparatus used is an ejector, similar in its general aspects to the well-known steam syphon bilge ejector, the only difference being that the present ejector is designed to operate with water pressure instead of steam. Its first permanent installation as a bilge pumping outfit was on the off-shore patrol boats of the Coast Guard, about three years ago where other means were not available. It was next installed on the Diesel — electric cutter *Northland*, and with such marked success that it was adopted for the new cutters.

The chief advantage of the system is that bilge piping, usually the most troublesome piping to keep up, is entirely eliminated. Two 4-inch ejectors are installed in the engine room, two in the fireroom, and one in each of the other water-tight compartments. Each ejector has its own strainer and foot valve and a direct overboard discharge pipe with a ball loaded valve at the ship's side. Pressure water is taken from the fire main through 2½-inch branches.

There is a similar installation of 2-inch ejectors throughout the ship taking pressure water from the sanitary system. These, because of their small size and low pressure, have a small capacity, but they are ample for the normal accumulation of bilge water.

This appears to be the simplest and most effective of all the ways of pumping bilges. In addition, there are no moving parts, there is no wear, any foreign object which will pass through the strainer will pass through the ejector, they cannot become air bound, and there are no valves except the foot valve and it is not essential to the perfect operation of the ejector.

The two fire pumps have a direct suction to the engine room bilge, so that, in case of a serious leak in the engine room, pressure water for the ejectors would also be taken from the bilge thereby materially increasing the capacity of the system. The ejectors are made by Schutte and Koerting, and are marketed under the atrocious name of "eductor."

Warren centrifugal pumps are used for the two fire pumps, each having a capacity of 250 gallons per minute against a pressure of 100 pounds per square inch. One pump is sufficient for five good streams through 7/8-inch nozzles, and at this capacity the pump discharge pressure is 70 pounds. One pump has a 30 H.P. D.C. motor, and the other a 30 H.P. Lee turbine. The pumps are identical.

There are three Warren centrifugal pumps used for a combination of sanitary, bilge ejectors, evaporator feed, ice machine cooling, auxiliary circulating, auxiliary air ejector, auxiliary lubricating oil cooling and deck washing. Each has a capacity of 200 gallons per minute against 25 pounds pressure. They take water from a header and discharge into a heater, so that one, two, or three pumps may be used, depending on requirements for low pressure water. Each is driven by a 5 H.P. motor, one being D.C. and two squirrel cage. One pump has an additional suction from the main condenser overboard discharge pipe, whereby hot water will be available for clearing the decks of ice.

Water is removed from the main condenser by a Westinghouse centrifugal pump driven by a 3 H.P. D.C. motor, and delivered through three passes of air ejector condensers to the filter tank. Air is removed from the main condenser by a duplex two-stage Westinghouse air ejector with inter- and after condensers. One first stage and one second stage is ample, leaving half the system in reserve. At full power the combined steam consumption on account of main circulating pump, main condensate pump and main air ejector is 614 pounds per hour and the vacuum ranges from 28½ to 29 inches.

There are two straight tube Babcock and Wilcox boilers, each with 2867 square feet of heating surface, and 333 square feet of interdeck superheater surface. The specified superheat is 250 degrees, and this with 250 pounds (gauge) pressure should give a steam temperature of about 650 degrees F. Actually it runs somewhat higher, averaging about 670 degrees. While engineers are agreed as to the need for high temperature steam, opinions are divided as to whether it should be obtained by high pressure saturated steam, or by low pressure superheated steam. Our shore-going brethren have now reached 1200 pounds pressure.

They use the steam part way through a turbine, take it out and use it for industrial purposes, then heat it up and put it back into the turbine further down. Such pressures and methods do not at present seem practicable or desirable for marine purposes. And while 250 pounds and 250 degrees does not utilize the utmost possibilities of steam, it does get most of them and it stays on this side of a good deal of trouble.

There are three burners on each boiler. They are the combined forced draft-natural draft burners made by the Babcock and Wilcox Company. With natural draft operation the boilers produced 17,000 pounds of steam of which 16,000 pounds had a total temperature of 584 degrees F. This is sufficient for a speed of 12.9 knots, and is ample for all ordinary cruising of the ship. Data for all conditions of operation have not yet been obtained, but it is estimated that the cruising radius should be 8000 miles.

Quimby screw pumps are used for fuel oil service. The speed of the motor driven pump cannot be regulated, and pressure is controlled either by a relief valve to the suction, or by bleeding through a return line to the bunkers. The other fuel oil pump is driven by a 5 H.P. Terry turbine.

Assuming that either the perfection of pulverized coal burning, or the scarcity of fuel oil, or both, might make a change of fuel desirable during the life of the ships, the fireroom is forward and adjacent to the oil tank bulkheads, and the port side contains fuel machinery only. Conversion to coal would involve no changes other than the installation of coaling chutes, bunker doors, and coal handling machinery.

The boilers are fitted with Diamond soot blowers and Babcock and Wilcox feed water regulators.

In hot weather in port a ship would be much more habitable if the boilers could be cooled off. But the small items of fire protection, bilge pumping, lighting and sanitation are sufficient to require that half the boiler plant be kept in operation unless they are otherwise provided for. For that reason each of these vessels has a Diesel-driven electric plant consisting of a 50 K.W. D.C. generator and a 30 K.W. A.C. generator direct connected to a four-cylinder Washington Estep engine. This set has no underway functions, but it holds considerable possibilities in comfort of

personnel, fuel saving, and cleanliness of boilers, for such time as the ship may be in port during warm weather.

These are the first Coast Guard vessels to be equipped with contra-propeller. The longitudinal component of the propeller stream is the part which is useful in driving the ship. Lateral components and rotary and swirling motions represent wasted effort. The name contra-propeller has been applied to certain stationary features which are designed to rectify the propeller stream and reduce the losses arising from rotation and turbulence. It catches the water as it leaves the propeller and changes the direction of flow in a method which is strongly suggestive of the stationary blading of a steam turbine. There are two vertical blades attached integrally to the rudder post, and two which are bolted to the rudder post and extend laterally in the same transverse plane. Much attention was also devoted to stream-lining the propeller hub and the rudder so that each detail is faired into the others.

The propeller is solid, 4-bladed bronze wheel, 11 feet 9 inches diameter, 11.07 feet pitch, and 61.1 square feet helicoidal area. On the ahead side the blades are planed to a true screw. It has the remarkably low apparent slip of 2 per cent at full power.

It is not possible to make a proper division of credit for the showing made as among hull design, propeller design, contra-propeller, and refinements of appendages; but the net result is that the ship has a speed about one knot greater than would ordinarily be expected for her dimensions, displacement and power.

The matter of weights is sometimes of vital importance; and the weights of the machinery required for an installation such as that described in this article are unquestionably much greater than those which can be had in other types of machinery. But machinery weights cannot be isolated,—they are tied up with total engineering tonnage which includes piping, manifolds, valves and fittings, boilers, fuel, reserve feed water and the water contained in the boilers, condensers and piping.

When working within fixed limits of engineering weights, power, and cruising radius, it is obviously permissible to increase the tonnage of machinery proper if thereby an equal tonnage is saved on other items.

FIG. 3.—AUXILIARY TURBO-MOTOR-GENERATOR.

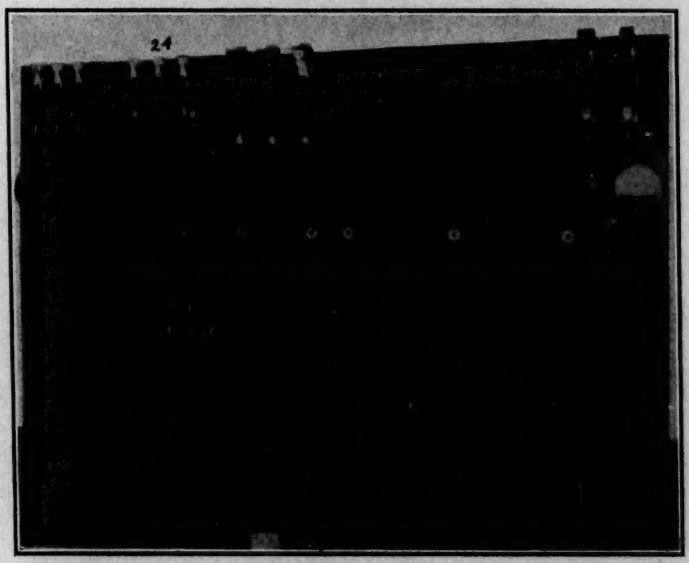
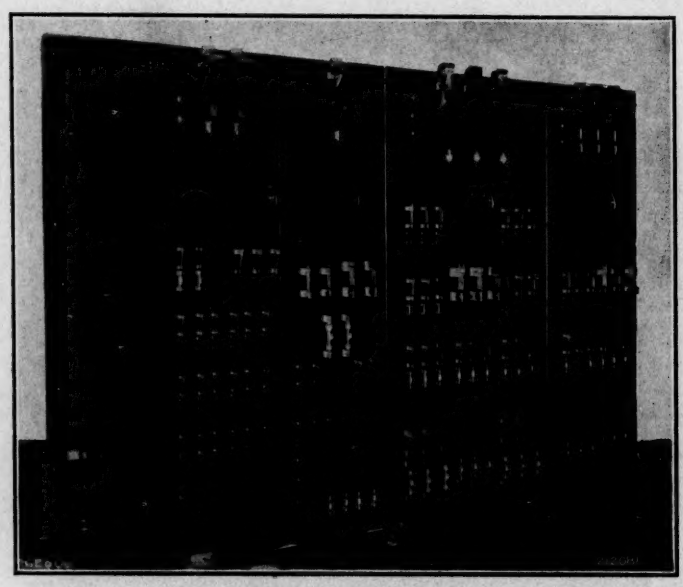


FIG. 4.—SWITCHBOARD.

I—1. 30 K.W.A.C. Generator Panel. Steering Gear Switch at Bottom.
I—2. Synchronizing Panel.
I—3. 100 K.W.A.C. Generator Panel.
I—4. 100 K.W.D.C. Generator Panel.
I—5. 100 K.W.D.C. Generator Panel.
I—6. 50 K.W.D.C. Generator Panel.



O-5 O-4 O-3 O-2 O-1

FIG. 5.—SWITCHBOARD.

O—1. A.C. Variable Frequency Panel.
O—2. A.C. Constant Frequency Panel.
O—3. D.C. Power Distribution Panel.
O—4. D.C. Lighting Distribution Panel.
O—5. Motor Ventilation Fan Motor Panel.

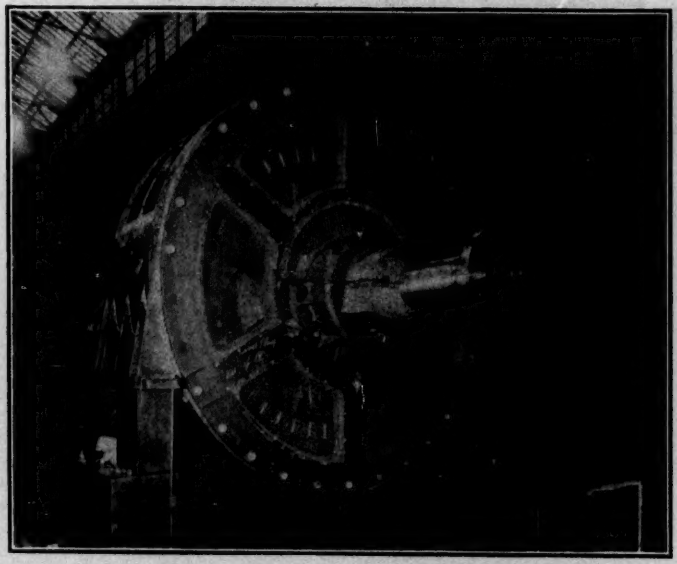


FIG. 12.—3000 H.P. SYNCHRONOUS PROPELLING MOTOR.

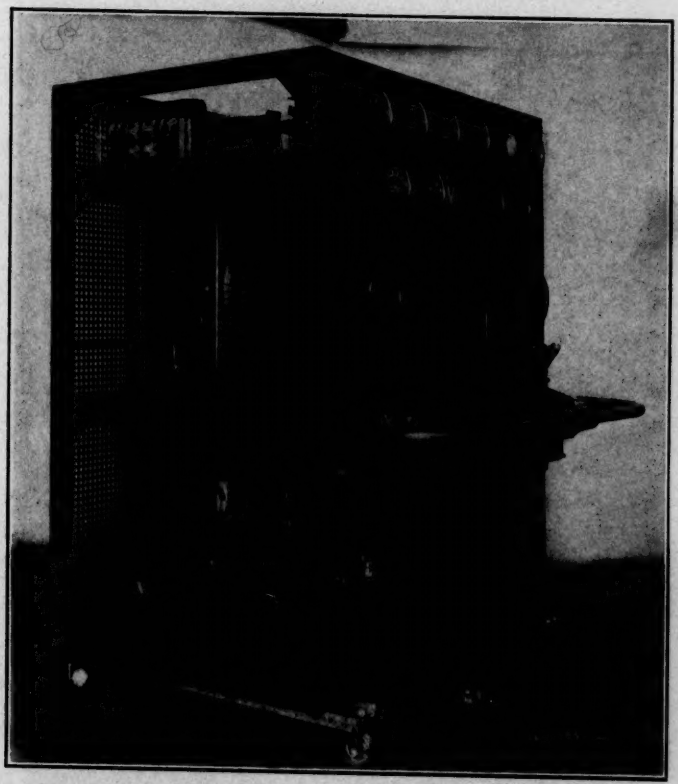


FIG. 11.—MAIN CONTROL PANEL. GRILLE-WORK PARTLY REMOVED.

For example, when the *Modoc* boilers, with a total of 6380 square feet of heating surface and superheating surface combined, were worked to their maximum output, the main motor developed 2656 shaft horsepower; whereas, on the *Pontchartrain* with the same kind of boilers, with a total of 6400 square feet of heating surface and superheating surface combined the main motor developed 3150 shaft horsepower without reaching the limit of boiler output. Thus, if weight had been an important consideration, the *Pontchartrain* boilers could have been designed for the *Modoc* rating, thereby saving considerable tonnage, of boilers and contained water.

The same argument applies with even greater force to the decreased tonnage of fuel for a given radius.

Experience to date indicates that the elimination of most of the piston rods and valve stems has greatly reduced the requirements for make up feed water, while the saving in weight of auxiliary steam and exhaust piping is substantial.

Practically the entire electrical equipment of these ships was purchased by the Coast Guard direct from the builders, the Westinghouse Electric and Manufacturing Company. The boilers were also purchased direct from the Babcock and Wilcox Company. It was necessary to run trials on one ship to ascertain whether these two contractors had met their respective guarantees. The *Pontchartrain* was selected for the test and advantage was taken of the opportunity to determine the complete performance of all the machinery. A few of the results, compared with results obtained on the *Modoc* trials, shows what may be obtained by the substitution of the central power station for the multiplicity of steam consumers.

It should also be observed that a real fuel saving is accomplished even when the central power station is only partly attained; which is the case when motorized auxiliaries are driven by an auxiliary turbine, whose economy is necessarily much lower than that of the main turbine.

There are two principal differences between the *Modoc* and the *Pontchartrain*, in addition to changes in types of auxiliary machinery; first, steam temperature on the *Pontchartrain* is about 200 degrees F. higher than on the *Modoc;* and second, the *Pontchartrain* is fitted with a contra-propeller.

In the appended comparison of the *Modoc* and the *Pontchartrain,* columns 1 and 2 show trial figures of the *Modoc* at full and three quarters speed, respectively; columns 3 and 4, the *Pontchartrain* at like speeds, auxiliaries driven by main generator; and columns 5 and 6, the *Pontchartrain* at like speeds, auxiliaries driven by auxiliary generator.

The full advantage of the central power station principle was not attained in the run shown in column 4, by reason of the fact that the full power nozzles of the main turbine were not tightly closed off. Unfortunately, the mistake was not detected in time to rectify it during the run.

	MODOC.		PONTCHARTRAIN			
Length overall,	240' - 00"		250' - 00"			
Length on water line,	220' - 00"		236' - 00"			
Beam, molded,	39' - 00"		42' - 00"			
Mean draught, trial, W.L.	14' - 03"		15' - 03½"			
Displacement, tons.	1660		1765			
Block coef., trial W.L.	0.480		0.470			
Prie. coef., trial W.L.	0.536		0.539			
Mid. sec. coef., trial W.L.	0.898		0.869			
Column No.	#1	#2	#3	#4	#5	#6
Steam at throttle, lbs., gauge	191	186	214	213	233	218
Steam at throttle, degrees F.	484	457	571	602	604	570
Back pressure, ins. mercury	2.15	1.48	1.31	1.3	1.01	1.0
S.H.P. main motor	2656	1008	2845	1155	3800	1197
S.H.P. main motor, % rated full power	102	38.8	94.7	38.5	126.7	39.8
S.H.P. main gen. to auxiliaries.	0	0	205	157	0	30
S.H.P. main gen. total	2656	1008	3050	1312	3800	1227
Speed, knots.	16.2	12.6	16.7	12.95	17.15	13.15
Speed, % rated full speed.	101	78.8	99.0	76.9	101.7	73.2
Fuel, lbs/hr. A.P.	3172	1549	2848	1271	2920	1416
Fuel, lbs/hr/S.H.P., A.P.	1.19	1.53	0.933	1.11	0.89	1.183
Live Steam, lbs/hr, main turbine	29520	10964	26187	14378	29520	12176
Aux. ex. to main turb. lbs/hr	+6700	+6705	-2530	-1435	-1618	-1128
Equiv. live steam, lbs/hr.	3830	2400	1140	515	615	405
Total equiv. live steam, lbs/hr	33350	13504	27257	14093	30045	12581
Equiv. steam to main turb. from and at 212	46200	23100	36900	19600	44250	21950

NOTES.

PURCHASE ON A WORK FACTOR BASIS.

Standard Tests for Industrial Supplies Bought by the United States Navy Are Applicable to Items Purchased by Manufacturing Plants.

By Timothy J. Keleher, Commander, U. S. N., and Frank M. McGeary, Material Engineer, U. S. N.

The phrase "purchase on a work factor basis" is used here to mean purchase on such a basis as will ensure the minimum cost per unit of service.

One way to determine differences in service is to test samples of the articles under consideration in a manner which simulates service conditions. For a person equipped with knowledge based on experience it is practicable to select the most important factors which enter into the service of articles, and in many cases it is possible, without prohibitive expense, to test these articles in a way that will ensure reasonably accurate results.

Specifications as commonly understood must of necessity be statements of minimum acceptable requirements. When competition becomes keen, this tends toward the delivery under contract of material which will be no better than the very lowest quality acceptable under the specifications.

QUALITY AND SERVICE VARY.

Materials of the same kind but of different manufacture will always vary as to quality. Hence if a way can be developed of determining these differences in quality, which means determining the differences in service to be expected, such development surely is desirable. It would appear reasonable, therefore, to work up specifications which would describe in detail the minimum acceptable requirements, and then to develop around each specification a work factor basis which would give the maximum service for the minimum cost.

At first thought, this may appear to involve a highly scientific investigation, during which cross currents may develop and lead in so many directions that no conclusive results will be possible. Further consideration, however, will show that it is entirely feasible for a consumer to develop, for many kinds of material, work factors based on actual service use of the material.

The problem is to select the variable factors entering into the service of a given material under its usual condition of use, and to reduce these factors to a simple mathematical formula, in conjunction with an easily duplicated test, which shall simulate—in a test laboratory, let us say—actual service conditions.

While it might be possible, in time, to evolve a scientifically correct work value for every brand of material, it is not believed that this is necessary

or even desirable, since a correction would have to be applied for each consumer, and for each use, to overcome differences in operating conditions. As the one who pays the bills, each consumer should decide what grade of material he desires to purchase, develop a specification as to minimum quality in order to identify the material, and establish work factors of a number of brands of the material, based on the kind of service expected. When the material is to be used for different kinds of service, the work value should be based on the most important service from the operating standpoint.

From a number of such studies, a few examples will be cited to illustrate what has been done along these lines and to point the way to future accomplishments.

HIGH PRESSURE ROD PACKING.

In the case of asbestos high pressure rod packing, comparative tests among 12 brands have shown the following results:

The quantities of packing required for the tests varied from 1.80 to 3.12 pounds.

The friction developed varied from 0.97 to 1.73 horsepower.

The endurance of packing varied from 371 to 2056 hours.

It is apparent from these results that there are at least three variables in the cost of rod packing for any stuffing box—weight of material necessary, friction developed, and endurance of the packing.

Examples of actual tests show that, assuming a uniform price of $0.80 per pound of packing, which under existing circumstances is a reasonable assumption, 68 per cent greater service should be expected from the best packing as compared with the poorest.

METHOD OF TESTING ROD PACKING.

The following method has been worked out for determining the work value of asbestos high pressure rod packings. A comparative test of a large number of brands of asbestos high pressure rod packing is now being made.

1. *Endurance Test, Reciprocating Rod.* The apparatus consists of a number of double-ended stuffing boxes mounted on two angle-iron frames, attached on opposite sides to a cast iron base. A steel rod 2 inches in diameter connected to a crosshead in each frame is made to reciprocate through the stuffing boxes. The crossheads in the angle-iron frames receive their motion from crank arms connected 180 degrees apart to a worm and gear on a 10-horsepower electric motor.

Each double-ended stuffing box is a cylindrical shaped casting 10 inches long, with a steam space in the center and a stuffing box at each end. The steam space of each casting is slightly larger in diameter than the stuffing boxes at each end, which are 3 1-16 inches in diameter and 3¼ inches deep.

By means of a valve mechanism, which receives its motion from a rocker arm connected to one of the crossheads of the apparatus, high pressure steam is alternately admitted and discharged to and from the steam space of the stuffing boxes during each stroke of the reciprocating rods. Throughout the test, the rods of the apparatus are operated at a speed of 100 strokes per minute with a stroke of 12 inches.

Two sets of packing are installed for test in the stuffing boxes of the apparatus. One set is installed in a lower or downward-facing stuffing box, and the other is installed in an upper or upward-facing stuffing box. During the test the weight and number of turns of packing initially and additionally installed in each stuffing box, and the gland adjustment necessary to prevent

or check steam leakage are recorded. The test is considered complete when leakage cannot be controlled. During the test the apparatus operates eight hours daily, being started cold each day.

2. *Friction Test, Reciprocating Rod.* This apparatus, with the exception of having only one double-ended stuffing box, is similar in operation and construction to the endurance-testing machine. The friction-testing apparatus is equipped with a stroke counter, a voltmeter, and an ammeter, which are used to determine the speed and the power of operation.

Before and after the test of each packing a zero load or the power required to operate the apparatus is determined. The weight, number of turns of packing installed in each stuffing box, and gland exposure are recorded before the test of each packing.

The readings of the stroke counter, voltmeter and ammeter are recorded at the start and hourly thereafter during the test. The gland exposures and the amount of steam leakage are also recorded during the test. The piston or reciprocating rod operates at a constant speed of 75 strokes per minute and with a stroke of 12 inches. The glands of the stuffing box are adjusted to compress packing so as to prevent or check steam leakages and absorb the least amount of power. Then if the glands are in a position where they cannot be further adjusted and the frictional load is still too high, the apparatus is stopped and a ring of packing is removed from each stuffing box. This test is continued for a period of 48 hours or more with not more than eight hours of operation daily, the apparatus being started cold each day.

3. *Friction Test, Rotary Rods.* The apparatus consists chiefly of a double-ended test stuffing box, a rotary steel rod, and a vacuum-obtaining device. The double-ended stuffing box is a cylindrical shaped casting 10 inches long, with a steam space in the center and a stuffing box at each end. The steam space is slightly larger in diameter than the stuffing box at each end. The stuffing boxes are 3¼ inches in diameter and 3¾ inches deep. The rotary rod is 2 inches in diameter and is made of cold rolled steel. It is connected directly to a high speed electric motor.

Samples of the packing to be tested are installed in each stuffing box of the test apparatus. For a given time during the test, the packing is subjected to working conditions, at steam pressure of 25 pounds gauge, to the highest obtainable vacuum, and to steam at a pressure of 150 pounds gauge. During the test the rod rotates at a speed of 3300 R.P.M., and the packing is subjected to 50 hours of running under each of the above conditions, as applicable. The glands of the stuffing boxes are adjusted to compress the packing sufficiently to prevent or to check leakage, and at the same time to absorb the least amount of power. Any uncontrollable steam leakage occurring during the test is recorded. During the vacuum run, any air that leaks through the packing into the steam space is drawn off through the vacuum obtaining device, passed through a meter, and measured.

The speed of the rod, and the power consumption due to frictional resistance are determined from hourly readings of the revolution counter, a voltmeter, and an ammeter. In order to determine accurately the power absorbed by the frictional resistance of the packing when subjected to the various conditions of the test, a zero load, or the power required to rotate the rod with no packing in the stuffing boxes, is determined before and after the test of packing.

The temperature and pressure of the steam in the steam space of the stuffing box are obtained from mercury glass thermometers and a calibrated compound gauge. The weight and number of turns of packing installed in each stuffing box, and the required gland adjustments are recorded during the test.

4. *Steam Conditions During Tests.* All high pressure rod packings are tested under saturated and superheated steam conditions as follows:

a. The endurance test reciprocating rod is run with superheated steam at pressures from 250 to 290 pounds gauge, and the steam temperature averaging 700 degrees F., with a maximum of 730 degrees F.

b. The friction test reciprocating rod is run with saturated steam at a pressure averaging 260 pounds gauge, with a maximum of 290 pounds gauge.

c. The rotary rod friction test is run with saturated steam at a maximum pressure of 25 pounds gauge.

5. *Method for Determining Work Value.* Upon completion of the test each packing is given a rating, corresponding to its relative work value. This is substantially a cost per hour of service determined by considering the following factors:

a. Total cost of packing required to complete the test (original installation plus additional rings required during the test).

b. Total effective service in hours.

c. Average friction (in horsepower) due to packing.

The cost per hour of service is determined by the following formula:

$$W = \frac{(b \times c) + (A \times d \times e)}{A}$$

in which

W = work value or cost per hour of service
A = total effective service, in hours
b = total weight of packing required for test in pounds
c = cost of packing per pound
d = average friction in horsepower
e = $0.01 per horsepower per hour (arbitrary cost of friction)

This formula is applied in Table 1, to a test of five samples of packing, and the work values shown in the last column are obtained.

TABLE 1. WORK VALUE OF ROD PACKING

Sample	Weight of Packing Required Lb.	Price per Lb.	Total Service Hr.	Friction Developed Hp.	Work Value Cost per Hr. of Service
A	2.84	$0.80	1421	0.97	$.0113
B	1.80	.80	2056	1.35	.0142
C	2.40	.80	371	1.30	.0162
D	3.12	.80	886	1.43	.0172
E	2.22	.80	1128	1.73	.0190

In the calculation used to arrive at the work value a price per pound of $0.80 has been used for all packings. The work value or cost per hour of service ranges from $0.0113 to $0.0190, the latter being 68 per cent greater than the former.

In Table 2 the work factor method is applied to purchases where different prices are bid on the brands represented by the samples.

FOUNDRY CRUCIBLES.

Foundry crucibles have been found to give erratic service. Manufacturers contend that crucibles are abused in service, while foundrymen claim poor results are due to inferior quality. There is probably some merit in both contentions, although under the system of purchasing crucibles in vogue heretofore, *i. e.*, award of contract to the low bidder, for crucibles conforming to the minimum requirements of the specifications, neither contention could be readily disproved. Under the work value method of purchase, it is possible to fix the responsibility for failure. If a manufacturer is found at fault, relief can be obtained in the form of additional crucibles, without cost to the consumer. If the foundrymen are at fault, steps can be taken to improve operating conditions in the foundry in question, and so indirectly to accomplish economies which, although not actually determinable, should be credited to the work factor system of purchase.

TABLE 2. SERVICE COST OF ROD PACKING

Sample	Weight of Packing Required Lb.	Bid Price per Lb.	Total Service Hr.	Friction Developed Hp.	Service Cost per Hr.
A	2.84	$0.68	1421	0.97	$0.0110
B	1.80	.78	2056	1.35	.0141
D	3.12	.85	886	1.43	.0173
E	2.22	.70	1128	1.73	.0186
C	2.40	.92	371	1.30	.0189

In an actual comparative test, it was found that the best set of samples of crucibles gave 102 per cent greater service than the next best set, and that the price of the best crucibles was 39 per cent greater than the price of the next best. Comparing the best with the poorest, the service difference was 586 per cent and the price difference 81 per cent.

METHOD OF TESTING CRUCIBLES.

The following method of obtaining work values has been established for foundry crucibles. Tests for approval of brand are made as follows: Three number 125 crucibles are used on regular foundry work, with the usual foundry practice, and in a natural draft coal or coke fire furnace. To make the tests as uniform as practicable, only copper-tin or copper-zinc-tin alloys containing less than 3 per cent lead and 0.05 per cent phosphorus are used. All test crucibles are used to destruction, and careful records are kept showing the number of heats, the number of hours in the fire, the composition of the material melted, from which the working temperature is computed, and the cause of failure. The work value is then determined by the simple formula:

$$\frac{a\ b\ c}{1000} \text{ in which}$$

a = average number of heats
b = average number of hours in the fire, per heat
c = average working temperature in degrees F.

In applying the formula to different sizes of crucibles, it was found that a better work factor was consistently obtained on smaller sizes than on larger sizes. This made it necessary to add a factor to the formula, which would reduce all sizes to a common basis. In actual practice, it has been found that the expression

$$\frac{\text{"average working temperature in deg. F."}}{1000}$$

added to the formula, gave a close check on different sizes of crucibles from the same manufacturing source.

A number of tests have been made on this basis in Table 3; two records are shown to give an idea of the results obtained.

From the calculation in Table 3, the crucibles represented by sample A should give 102 per cent more service than those represented by sample B.

TABLE 3. FOUNDRY CRUCIBLE TESTS

Record of Tests

Item	Crucible No. 1	Crucible No. 2	Crucible No. 3
Sample A			
Average no. of heats *a*.....	76.00	76.00	86.00
Av. no. hr. in fire *b*.....	2.97	3.47	3.34
Av. temp. in deg. F. *c*.....	2080.00	2070.00	2060.00
Sample B			
Average no. of heats *a*.....	43.00	35.00	45.00
Av. no. hr. in fire *b*.....	3.21	3.21	3.02
Av. temp. in deg. F *c*.....	2050.00	2050.00	2050.00

Results

Item	Sample A	Sample B
Crucible No. 1....	$\frac{76 \times 2.97 \times 2080}{1000} = 467.92$	$\frac{43 \times 3.21 \times 2050}{1000} = 282.96$
Crucible No. 2....	$\frac{76 \times 3.47 \times 2070}{1000} = 545.89$	$\frac{35 \times 3.21 \times 2050}{1000} = 230.31$
Crucible No. 3....	$\frac{86 \times 3.34 \times 2060}{1000} = 590.85$	$\frac{45 \times 3.02 \times 2050}{1000} = 278.59$
Total.....	1604.66	791.86
Work value (average)	534.89	263.95

To check deliveries, samples are taken from each lot of crucibles delivered on contract, and tested in the receiving foundry, by the method described.

When crucible contracts provide for deliveries at a number of foundries belonging to the same organization, the check test will serve two purposes: it will determine whether crucibles of uniform quality are supplied on contract, and it will give a direct comparison of the results obtained at the different foundries.

Applying the work factor to purchases, the results shown in Table 4 are obtained.

TABLE 4. SERVICE COST OF CRUCIBLES

Bid	*a* Bid Price per Crucible	*b* Average No. Heats	*c* Average No. Hr. in Fire	*d* Average Working Temp. Deg. F.	*e* Work Factor	Service Cost of Crucible per Hr. *a* ÷ *e*
A	$10.095	79.3	3.256	2070	534.89	$0.0204
B	7.184	41.0	3.146	2050	263.95	0.0268
C	7.184	21.0	3.300	2060	142.76	0.0503
D	7.256	22.0	3.070	2050	138.45	0.0523
E	7.980	20.0	3.220	2060	132.66	0 0603
F	5.565	11.3	3.400	2030	77.99	0.0713

Like up-to-date manufacturing plants, the United States Navy maintains a strict watch over the quality and properties of all materials and supplies which it buys. In order to hold to the standards set, and to treat all suppliers impartially, solely on the merits of the goods they furnish, the Navy Bureau of Engineering has worked out methods of specifying and testing which will undoubtedly be of considerable use to manufacturing plants. Whether a concern furnishes materials and supplies to the Navy and hence wants to know the requirements of this department, or whether it wishes to apply equitable and adequate methods to its own purchases, the average manufacturing plant will undoubtedly find the procedures described and the results obtained of interest and utility.

The idea back of the entire plan is to place purchasing on a practical work factor basis. It is not easy to develop test methods which will measure the performance of materials and equipment according to the way they are to be applied in actual service. But service conditions are the only real criterions by which to judge the relative merits of a number of similar products. Ordinary examination and analysis may indicate that an article on the market has the quality and characteristics that make it worth the price for the purpose it is to serve. When used, however, it may develop flaws and weaknesses which render it valueless. It was to replace inadequate testing procedures with those that will show up quality or failure under conditions as close as possible to actual service that the present methods were evolved.

MINERAL LUBRICATING OIL.

Beginning with everyday supplies, the Navy Department Bureau of Engineering decided to study the application of work factors to mineral lubricating oil, probably the most commonly used material and at the same time perhaps the most complex.

In approaching the subject, one marvels at the number of different kinds or brands of lubricating oil on the market for almost every kind of machine. Such variety creates the impression that each is fundamentally different from all the others. While this condition may exist, it is not necessarily true, nor does it follow that one kind of oil will not serve satisfactorily for a number of similar purposes.

Through establishing certain minimum requirements which oils must meet, it is frequently held possible to make up a list of oils such that—assuming the selection of the proper viscosity—any oil on the list will meet a specific requirement, as, for example, to provide satisfactory lubrication in any forced-feed or motor-cylinder lubricating system. These "good" oils, however, will vary somewhat, though not so much in the character of the service rendered as in the length of time during which they will maintain the initially good service.

The real purpose of lubricating oil is to form a film which shall separate rubbing surfaces so as to prevent metallic contact, and thus reduce friction, and in some cases to form a seal. Both bearing pressures and temperatures govern the viscosity of the oil to be used, and, considering well refined oils of good basic quality, this viscosity need be the only factor considered in selecting a suitable lubricant for machinery. All other differences among oils of the same viscosity, for a number of purposes, are considered to be questions of refinement or purity.

The highest degree of refinement or purity is required for steam turbine lubrication, a lesser degree may be used for internal combustion engines, while for railroad car axles oil of little refinement will answer. Oil suitable for turbine lubrication may be used for railroad car axle lubrication, but as a general proposition it will be a more expensive lubricant than is required for the purpose.

To establish a work factor for lubricating oil, one should start with an oil of the highest degree of refinement or purity and assign to that oil a work value of 1.00. All oils of lesser degrees of refinement will then have work values of less than 1.00.

To determine the work value for mineral lubricating oils, the best method is believed to be to consider only the chemical and physical properties affected by use, and to grade the oils on the *changes taking place during use* under actual operating conditions, or preferably under conditions which simulate operating conditions as to bearing pressures and temperatures, and from which all chance for dilution and contamination is removed. Such a test would show the *changes taking place due to wear*, which changes are those which affect the value of the oil.

A machine similar to the one here described is suited for this test:

It consists of a test journal of steel, 3 3-16 inches in diameter and 6 inches long, revolving in a babbitted bearing and driven by a variable speed motor, calibrated for foot-pounds lost work at different speeds. Force feed lubrication is used. The lubricating oil is circulated from the reservoir to the top on-side of the bearing by means of a motor-driven rotary pump. A pressure of 150 pounds per square inch is uniformly applied to the test journal by means of four calibrated springs, each provided with a scale. The temperature of the bearing is obtained from readings of two mercury-glass thermometers, which are set in two thermometer pockets bored in the upper part of the bearing. The lubricating oil is neither cooled nor heated, but allowed to attain any temperature which it will during the test. The motor driving the test journal having been calibrated, the input power to the motor is obtained from calibrated voltmeters and ammeters.

The endurance test should be continued for 100 hours at the rate of 8 hours per day, starting cold each day. At its conclusion a comparison of the physical and chemical properties of the oil with those obtained before the test began should enable one to decide as to the relative value of two different oils. The assignment of certain limiting values (minimum requirements given as parts of specifications) to each of the variable factors should then enable one to grade a number of lubricating oils by a mathematical scale. This has been done in the following paragraphs dealing with the factors which change during use under the conditions of the endurance test. These factors are:

a. Viscosity
b. Organic acidity as acid number
c. Matter insoluble in 88 degrees Baumé gasoline
d. Carbon residue

All other chemical and physical properties have been disregarded for the reason that the factors mentioned above are the only ones in which changes during the endurance test should be tolerated.

Fluidity and its accurate measurement have been the subject of recent scientific study which promises a solution in the near future and may prove fluidity to be a factor which must, in the nature of things, be given consideration. It is thought that present knowledge of the subject does not warrant its inclusion at this time.

All other factors will have been considered in the test for physical and chemical properties of the oil prior to use. The bearing temperatures and the coefficient of friction have also been disregarded, since these will depend to a very large extent upon the changes occurring in the variable factors listed above, and also because it is believed best to take cognizance only of changes in the oil due to wear produced by work done on it.

Having selected the factors for consideration, it is necessary to decide how they should be considered. The following procedure should be carried out:

A. Viscosity is usually determined at three temperatures, *viz:* 100, 130 and 210 degrees F. Examination of a large number of tests of lubricating oils which have been proved satisfactory in service, show the increase in viscosity at the temperatures mentioned to be within a limit of 20 per cent after the endurance test. Therefore, an increase in viscosity up to 20 per cent at one or all of the three temperatures is considered to be acceptable. An increase of more than 20 per cent at any of the three temperatures is considered to be not acceptable.

Reducing increase in viscosity to a work factor basis, zero increase would warrant a work value of 1.00 and an increase between zero and 20 per cent, some value less than 1.00. The following formula will be used to compute the work value of increase in viscosity at each temperature, three work values entering into the final work factor:

$$\text{Work value} = 1.00 - \frac{\text{Final viscosity} - \text{Initial viscosity}}{\text{Initial viscosity}} \Big/ 0.20$$

The following viscosity data was obtained on an aviation lubricating oil before and after a 100-hour test on the endurance machine:

TEMPERATURE	VISCOSITY INITIAL	VISCOSITY FINAL
100°F.	1014	1125
130°F.	379	414
210°F.	78	83

Applying the formula we get the following:

$$\text{(at 100°F.)} \quad 1.00 \quad \frac{1125 - 1014}{0.20 \times 1014} = 0.4528$$

$$\text{(at 130°F.)} \quad 1.00 \quad \frac{414 - 379}{0.20 \times 379} = 0.5328$$

$$\text{(at 210°F.)} \quad 1.00 \quad \frac{83 - 79}{0.20 \times 78} = 0.7469$$

B. Organic acidity, matter insoluble in 88 degrees Baumé gasoline, and carbon residue will all increase in content to some extent during the endurance test, the poorer oils showing the greater increases. From a study of a large number of tests of lubricating oils, it has been found that in practice the best results are obtained when the initial content and the increase in content during the endurance test are low. Table 5 shows the initial accept-

TABLE 5 TOLERANCE LIMITS OF OILS

Factor	Maximum Content Initially Acceptable	Initial Rejection Point	Final Rejection Point
Organic acidity as acid number	0.10	0.11	1.01
Matter insoluble in 88 degree Baumé gasoline, per cent	None	0.05	10.10
Carbon residue—light, medium or heavy, per cent	0.50	0.55	1.05
Carbon residue, extra heavy or ultra heavy, per cent	1.00	1.05	2.05
Carbon residue, aviation, per cent	1.50	1.55	3.05

able content, the initial rejection content, and the final rejection content for the factors mentioned.

Inasmuch as any content beyond that allowable is considered sufficient reason for rejection of an oil, two ratings will be given to each of these factors; one based on the initial content, and the other on the increase in content during the endurance test. Two formulas as follows will be used:

$$1. \quad \frac{\text{Initial rejection point} - \text{Initial content}}{\text{Initial rejection point}} = \text{Work value No. 1}$$

$$2.\ \frac{\text{Final rejection point} - \text{Final content}}{\text{Final rejection point} - \text{Initial content}} = \text{Work value No. 2}$$

The chemical properites in Table 6 were obtained on the same aviation lubricating oil as that mentioned above, before and after the endurance test: Applying formulas No. 1 and No. 2 to each we get the following:

TABLE 6
CHEMICAL PROPERTIES OF AVIATION OIL BEFORE AND AFTER USE

Factor	Before Use	After Use
Organic acidity number	0.08	0.32
Matter insoluble in 88 degree Baumé gasoline, per cent	None	0.50
Carbon residue, per cent	0.87	1.13

A. Organic acidity

$$1.\ \frac{0.11 - 0.08}{0.11} = \frac{0.03}{0.11} = 0.2727$$

$$2.\ \frac{1.01 - 0.32}{1.01 - 0.08} = \frac{0.69}{0.93} = 0.7419$$

B. Matter insoluble in 88 degrees Baumé gasoline

$$1.\ \frac{0.05 - 0.00}{0.05} = \frac{0.05}{0.05} = 1.000$$

$$2.\ \frac{10.10 - 0.50}{10.10 - 0.00} = \frac{9.60}{10.10} = 0.9504$$

C. Carbon Residue

$$1.\ \frac{1.55 - 0.87}{1.55} = \frac{0.68}{1.55} = 0.4387$$

$$2.\ \frac{3.05 - 1.13}{3.05 - 0.87} = \frac{1.92}{2.18} = 0.8807$$

TABLE 7 RESULTS OF TESTS ON TWO CLASSES OF OILS

	Light (100-115 at 130 Degrees F.)						Ultra Heavy (300-320 at 130 Degrees F.)					
	L 1		L 2		L 3		U 1		U 2		U 3	
	Before	After	Before	After	Before	After	Before	After	Before	After	Before	After
Chemical Properties:												
Mineral acids and free alkali	None	None	None	None	None	None	None	None	None	None	None	None
Organic acidity as acid No.	0.05	0.32	0.00	0.80	0.06	0.15	0.05	0.42	0.06	0.18	0.05	0.76
Matter insoluble in 86 deg. Baumé gasoline	None	2.00	None	1.00	None	None	None	5.00	None	0.50	None	10.00
Free sulphur or corrosive sulphur compounds	None	None	None	None	None	None	None	None	None	None	None	None
SO_3 as sulphonates	None	*	None	*	None	*	None	*	None	*	None	*
Total sulphur, per cent	0.41	*	0.32	*	0.22	*	0.18	*	0.20	*	0.32	*
Soluble in cold concentrated sulphuric acid, per cent	10.00	*	5.00	*	10.00	*	8.00	*	9.00	*	21.00	*
Water, per cent	None	*	None	*	None	*	None	*	None	*	None	*
Emulsifying constituents with steam	None	Present	None	None	None	None	None	None	None	Present	None	None
Carbon residue, per cent	0.07	0.44	0.03	0.06	0.04	0.09	0.02	0.15	0.33	0.60	0.07	0.67
Nature of carbon residue	Stain	Hard	Stain	Loose	Stain	Stain	Stain	Loose	Loose	Loose	Loose	Loose
Ash, per cent	None	None	None	None	None	None	None	None	None	None	None	None
Evaporation loss, per cent (50 grams of oil heated for 7 hrs. at 356 deg. F.)	0.23	*	0.12	*	0.04	*	0.48	*	0.34	*	0.08	*
Saponifiable oils	None	*	None	*	None	*	None	*	None	*	None	*
Physical Constants:												
Specific gravity at 60-60 deg. F.	0.903	0.905	0.892	0.898	0.890	0.892	0.955	0.954	0.915	0.917	0.938	0.940
Baumé degree	25.20	24.85	26.95	26.95	27.30	25.90	16.60	16.75	23.00	22.67	19.25	18.94
Flash point, O. C., deg. F	385	390	400	400	405	405	385	390	425	430	400	405
Fire point, O. C., deg. F	455	450	440	450	445	445	415	410	470	475	465	465
Pour point, deg. F	30	30	25	*	15	15	15	15	5	10	15	15
Viscosity, Saybolt Universal												
at 100 deg. F. seconds	193	213	201	213	209	209	1050	1160	795	881	945	1097
at 130 deg. F. seconds	100	107	105	111	104	105	348	347	322	336	328	375
at 210 deg. F. seconds	45	47	45	46	47	47	56	58	70	73	63	72
Work Values:												
Viscosity at 100 deg. F.	.....	0.3520	.....	0.7015	.....	0.7750	.....	0.4785	.....	0.4590	.....	0.1940
Viscosity at 130 deg. F.	.....	0.6800	.....	0.7143	.....	0.9519	.....	1.0000	.....	0.4375	.....	0.2535
Viscosity at 210 deg. F.	.....	0.7872	.....	0.8889	.....	1.0000	.....	0.8215	.....	0.7857	.....	0.3455
Organic acidity ..... before	.....	0.5454	.....	0.3388	.....	0.5385	.....	0.5454	.....	0.5318	.....	0.5454
Organic acidity ..... after	.....	0.7083	.....	0.4425	.....	0.9149	.....	0.6146	.....	0.8737	.....	0.2604
Matter insoluble 86 deg. Baumé gasoline ..... before	.....	1.0000	.....	1.0000	.....	1.0000	.....	1.0000	.....	1.0000	.....	1.0000
Matter insoluble 86 deg. Baumé gasoline ..... after	.....	0.8019	.....	0.5000	.....	1.0000	.....	0.9804	.....	0.9804	.....	0.9020
Carbon residue ..... before	.....	0.8726	.....	0.9454	.....	0.9272	.....	0.8455	.....	0.8430	.....	0.6669
Carbon residue ..... after	.....	0.6234	.....	0.9706	.....	0.9544	.....	0.8000	.....	0.8718	.....	0.2100
TOTAL	.....	6.4696	.....	7.2696	.....	8.0802	.....	6.8097	.....	8.5718	.....	4.2100
Average or work factor	.....	0.7188	.....	0.8077	.....	0.8946	.....	0.7566	.....	0.7302	.....	0.4677

*Not determined.

Collecting the work values for the different factors, and striking an average, we have a "work factor" for the oil as follows:

Viscosity at 100 degrees F	0.4528
Viscosity at 130 degrees F	0.5528
Viscosity at 210 degrees F	0.7469
Organic acidity (A-1)	0.2727
Organic acidity (A-2)	0.7419
Insoluble matter (B-1)	1.0000
Insoluble matter (B-2)	0.9504
Carbon residue (C-1)	0.4387
Carbon residue (C-2)	0.8807
Total	6.0369
Average	0.6708

Referring to Table 7, which is devoted to light and ultra heavy oils, application of the work factors to purchases would be as shown in Table 8.

TABLE 8 SERVICE COST OF OILS

Oil	A Bid Per Gallon	B Work Factor	C A+B Service Cost Per Gal.
L 1	$0.28	0.7188	$0.3895
L 2	0.30	0.8111	0.3698
L 3	0.33	0.8946	0.3688
U 1	0.48	0.8205	0.5850
U 2	0.46	0.7302	0.6297
U 3	0.38	0.4677	0.8125

It should be noted that the prices given above are those which are obtainable on purchases of lubricating oil which aggregate 1,500,000 gallons per year.

[Other tables, supplementary to Table 7 and containing equally complete data, have been prepared by the Navy Department for oils graded as medium, heavy, extra heavy, winter, summer, and Grade 4.]

To obtain the materials and supplies that best meet service requirements the Bureau of Engineering of the United States Navy has instituted new testing methods under which to make purchases. These methods have been developed to reproduce conditions of actual use as closely as possible.

Usual forms of examination and analysis may pass materials which, while meeting specifications, do not yield the service offered by other commodities. Unless thorough tests are carried on to bring out the true economic ratio between two or more similar materials, the temptation prevails to buy the lowest in price, provided it comes within the specifications. But the lowest priced commodity in actual service may not approach the degree of efficiency shown by higher priced items. Something costing twice as much may give three times the service.

The methods described in this article were developed to place purchases upon a work factor basis instead of upon a price basis, and to establish accurate and impartial means of testing. They are of interest not only to

manufacturers furnishing supplies to the Navy Department, but also to all plants desiring to institute an effective program for measuring the actual utility of commodities bought on the market under competitive bids.

VALVE-GRINDING COMPOUND.

On a test made by the Navy Bureau of Engineering on seven samples of valve-grinding compound, "pounds ground away per pound of valve-grinding compound" varied between 0.173 and 0.403. The best compound gave over 2.3 times greater service than the poorest, whereas the price of the best compound was only 1.17 times that of the poorest. The best removed a pound of metal in 17 hours, while it took forty hours to remove a pound of metal with the poorest compound. The labor costs with the poorest compound would be about 2.3 times as much as when using the best.

The following method has been developed for determining the work value of valve-grinding compounds. The test is made on a machine designed to simulate the work of grinding an engine poppet valve in its seat. The machine consists of three units with a single driving mechanism. The valve seat is simulated by a 2-inch diameter, ½-inch thick disc of cast iron of 143 Brinnell hardness number, which is superimposed on a seat which simulates a valve, being steel of 267 Brinnell hardness number.

While these valves vary considerably from actual conditions sometimes found, the important thing is to have all compounds tested identically, and these values are maintained throughout each test. The shaft on which the disc is attached is rotated 1¼ revolutions in one direction by a rack operating a pinion, then driven a similar number of revolutions in the opposite direction. Once during each such cycle the disc is lifted completely from its seat by a cam attached to the rack. The machine is driven at a uniform speed of 70 cycles per minute. While seated, the discs are pressed together with a pressure of 5 pounds per square inch by means of the top weight. Grinding compound is fed to the surface between the discs through a groove on the seat from graduated compression cups turned by hand every 2 minutes at such a rate as to give 6.5 grams of compound to the surface every hour. Each top and bottom disc is cleaned and weighed on chemical balances before and after each 2-hour test.

A work value for each grade is determined as follows:

$$\frac{W_1 + W_2}{453.59} \div \frac{A}{453.59} = \text{pounds of material ground away per pound of grinding compound used,}$$

in which

W_1 = weight in grams ground away from valve
W_2 = weight in grams ground away from seat
A = total weight of compound used in grams
453.59 = number of grams in a pound.

Table 9 shows the results obtained by applying this formula on a test of grinding compounds.

Table 10 summarizes the results obtained by applying the work factor to purchase. The product of bidder G costs, per pound of grinding compound, 1.17 times that of bidder I, but the cost of removing a pound of metal with the "G" compound is approximately $0.74 as compaeed with $1.47 to remove a pound of metal with the "I" compound. This does not take into consideration differences in the cost of labor as represented by column *b*, which gives the pounds of metal removed per hour.

HACK-SAW BLADES.

On a test of thirteen samples of hack-saw blades for hand frames, the number of cuts per blade varied between 8.33 and 22.33, and the number of strokes per cut varied between 4602 and 8523. The best set of samples gave 167 per cent greater service than the poorest set, whereas the price of the best blades was only 10 per cent greater than the price of the poorest. A similar condition existed in the hack-saw blades for machine use.

TABLE 9. TEST OF GRINDING COMPOUNDS

Exhibitor	Grade	Grams Ground Away		Work Factor
		Top	Bottom	
G	Fine	1.40	3.10	0.346
G	Coarse	2.90	3.10	0.461
H	Fine	1.10	1.20	0.177
H	Coarse	1.70	1.50	0.246
I	Fine	1.30	0.80	0.161
I	Coarse	1.50	0.90	0.184

TESTING HACK-SAW BLADES.

In making the tests, hand frame blades were used in a special machine, and machine blades were used in another machine suitable for the heavier kind of work. Discs 3 inches in diameter were cut from annealed tool steel of the following chemical analysis:

Carbon	0.81 to 0.90 per cent
Manganese	0.15 to 0.35 per cent
Phosphorus	0.02 per cent, maximum
Silicon	0.10 to 0.40 per cent
Sulphur	0.03 per cent, maximum

TABLE 10. WORK FACTOR APPLICATION

Bidder	Grade	*a* Bid Price Per Lb.	*b* Work Factor	Cost Per Lb. of Metal Removed Col. *a*/*b*	Average
G	Fine	$0.2950	0.346	$0.8526	$0.7462
G	Coarse	0.2950	0.461	0.6399	
H	Fine	0.2650	0.177	1.4971	1.2871
H	Coarse	0.2650	0.246	1.0772	
I	Fine	0.2523	0.161	1.5670	1.4691
I	Coarse	0.2523	0.184	1.3712	

Each blade was used until unfit for further use. The failure of the blade was marked by breakage, cutting at an angle, or cessation of cutting. Both machines were operated at 60 strokes per minute, the speed and pressure being constant during the test. In Table 11 the data for three blades of identical description illustrates the method of computing the work value of each type and size of blade:

TABLE 11. HACK SAW BLADE TESTS

Blade No.	Cuts Made Before Failure	Average Strokes per Cut
1	14	6,000
2	20	8,000
3	11	4,000
	45	18,000
Average	15 Gen. Avg.	6,000

The following formula is used:

$$\text{Work value} = \frac{\text{Average cuts per blade}}{\text{General average strokes per cut}} \times 10{,}000, \text{ or in the example given,}$$

$$\frac{15}{6000} \times 10{,}000 = 25$$

Numerous tests were made on this basis. Five records are shown in Table 12 to give an idea of the results obtained.

TABLE 12. WORK VALUE OF BLADES

Tests	Cuts per Blade	Strokes per Cut	Work Value
A	17.33	4,602	37.7
B	22.33	7,560	29.5
C	16.33	6,412	25.5
D	16.00	8,523	18.8
E	8.33	5,912	14.1

Sample A gave 167 per cent greater work value than sample E.—"Manufacturing Industries."

SURFACE CONDENSER TESTS.

Now that every station engineer is anxious to secure the last 1-10 of an inch vacuum commercially possible, it is important that tests of surface condensers should be comparable *inter se*, the more especially, in that these highly-important accessories to the steam turbine are often supplied by other contractors than those responsible for the prime mover. It should be possible to appraise the relative merits of condensers operating under different conditions, as easily as it is to determine the relative efficiencies of two steam turbines. In the latter case, it has long been recognized that the efficiency ratio taken *per se* may be most misleading. A high ratio may, in fact, often be attributable rather to the boiler and superheating plant than to the skill and knowledge of the turbine designer. Under ordinary power-house conditions, an increase of the superheat from 100 degrees to 300 degrees F. will improve the efficiency ratio of the turbine by from 3 per cent to 3½ per cent. There are, of course, methods by which results obtained with different superheats can be reduced to a common basis, and the "hydraulic efficiency" of the turbine computed.

No similar method has yet been devised for reducing to a common denominator the results of surface condenser tests. The relative merit is quite commonly assumed to be given by "K," which is defined as the heat transmitted per square foot of cooling surface per hour and per degree of mean difference of temperature between the steam and the water. Such a criterion ignores the fact that the resistance to heat transfer between the tube and the circulating water is not independent of either the temperature of the water or of its velocity.

The defects of "K" as a standard of comparison are, moreover, aggravated by the methods by which it is computed.

In some cases, for instance, a conventional value has been assumed for the total heat of the steam entering the condenser. While turbine efficiencies ranged low, it was permissible, at least for purposes of design, if not for those of comparison, to assume that some, say, 1050 B.T.U. were carried into the condenser by each pound of exhaust steam. Under modern conditions, however, this figure may be a serious overestimate, and any conventional figure is quite out of place in condenser tests, the more especially since it is only recently that provision has been made for direct measurement of the supply of circulating water. Lacking this, it is necessary to deduce the quantity of circulating water passing through the condenser from its temperature rise and from the weight of condensate collected. Under modern conditions, the total heat of the steam entering the condenser may be as little as 950 B.T.U. per pound, while in certain condenser tests made, some 15 or 16 years ago, the corresponding figure ranged from 1044 B.T.U. at full load to 1059 at half load. It is obvious, therefore, that conventional figures for the total heat of exhaust steam are entirely unjustifiable, and this is duly recognized in the code for the conduct of surface condenser tests, which was published in 1927 by the Heat Engine Trials Committee appointed by the Institution of Civil Engineers. This code is, however, somewhat seriously defective as regards one or two items, and an early opportunity should be sought for revising and amending the faulty paragraphs.

Even when the heat content of the exhaust steam is accurately determined, the value of "K" is still very far from satisfactory, as a sole basis for the comparison of condensers tested under different conditions. With a mean water temperature of 60 degrees F., the resistance to a transfer of heat from the inner surface of the tubes to the circulating water is (in normal conditions) some 20 per cent higher than when the mean water temperature

is 85 degrees F. With modern methods of eliminating air, and with modern methods of tube cleaning, this resistance is nearly 80 per cent of the total opposing the flow of heat from the steam to the cooling water. The resistance also varies rapidly with the water speed, but this important factor is entirely ignored in the computation of "K."

With the dirty tubes, and extravagant air leakage, characteristic of condensers in pre-turbine days, it may have been legitimate to take as constant the total resistance between steam and water. This assumption is the basis of the logarithmic formula for mean temperature difference, which is that recommended for use by the Heat Engine Trials Committee. In modern conditions, this procedure very seriously underestimates the true mean temperature difference, and is responsible for values of "K" approaching 1000 B.T.U. per square foot per hour per degree of mean temperature difference. It may safely be said that while this figure is not impossible, it is at least highly improbable that it has ever been achieved in commercial condenser practice. Thus, in certain careful tests made some fifteen or sixteen years ago at a north country power station, the heat flow was 7090 B.T.U. per square foot per hour. The circulating water had a temperature of 81 degrees F. at inlet and of 94 degrees F. at discharge, while the steam temperature was 96.6 degrees F. The logarithmic mean temperature difference was thus about 7.26 degrees, giving for K a value of 977 B.T.U. per square foot per hour per degree of mean temperature difference. The true value of the mean temperature differenc was, however, probably of the order of 9 degrees F, giving 788 as the corresponding value of K. This is, of course, still an excellent result, being the best we have yet met with, but as already mentioned, the coefficient K, even when accurately determined, is far from satisfactory *per se* as a basis for the comparison of tests.

It will be noted, that in this test temperatures were measured only to the nearest degree. This is inadequate under modern conditions, and the Heat Engine Trials Committee's code quite properly provides for readings to 1-10 degree F.

Striking confirmation of the fact that the logarithmic formula may lead to very erroneous estimates of the mean temperature difference are provided by Mr. Selvey's tests* of the surface condensers at the Valley-road power station, Bradford. In these tests the temperature of the circulating water was measured not only at inlet and outlet, but also between the two passes. It is thus possible to determine the mean water temperature by Cotes' well-known formula, which may be written as $t = \frac{1}{6}(t_0 + t_1) + \frac{2}{3} t_{\frac{1}{2}}$, where t_0 and t_1 denote the temperatures at inlet and discharge respectively, and $t_{\frac{1}{2}}$ the temperature at the mid-point of the water path.

In almost every test the mean temperature difference determined in this way was very materially larger than that deduced from the logarithmic formula. Thus, for example, in test No. 10 the value of K is 637, as estimated by the logarithmic formula, and 591 as calculated from the mean water temperature.

Provision is made by the Heat Engine Trials Committee for taking the temperature of the circulating water after each pass, but there is no proposal to make any use of the observations. A note is, in fact, added to the effect that such measurements may be unreliable owing to the existence of stratified flow, and it may well be that the mid-point temperatures recorded in the Valley-road tests are a little too high and that the logarithmic formula is even more seriously in error than is indicated by the figures cited above.

*See "Engineering," vol. cxxiii, page 161 (1927).

The obvious remedy for the possibility of error, to which the committee call attention, is to use a sufficiency of thermometers suitably distributed, so as to ensure that the average temperature of the water, between the passes, is truly determined. If this be done, then the mean water temperature can be found with all the precision necessary by means of Cotes' rule, or by an appropriate modification thereof should the tubing not be equally divided between the two passes. Under modern conditions the mean temperature difference is so small that quite small errors in this figure may make very material errors in the value of K.

There is another point in the code, which also seems open to criticism, *viz.*, no definite recommendation is made as to how the steam temperature required in computing K is to be determined. Provision is made for observing both temperatures and pressures at the exhaust flange. The temperature thus observed is commonly less than that corresponding to the pressure, and with the small mean temperature differences now usual, the value of K will differ widely according as one or the other steam temperature is employed. In view of the fact that the condenser tubes are covered with a film of moisture having an average thickness of the order of 1-700 inch, and thus many molecules thick, it is obvious that the temperature used in computing K should be that corresponding to the observed pressure. It is possible, however, that there is room for improvement in the methods of determining the latter in order to eliminate more adequately the effects of eddies and the like.

In conclusion, attention may be drawn to the remarkable fact that under modern conditions the vacuum attained in the London area is in some cases higher in cooling tower stations than it is in others more favorably situated, and having the advantage of practically unlimited supplies of cooling water. This is in part due to the greater readiness of power station engineers nowadays to spend money on their condensing plant. Much more cooling surface is now provided than was formerly the case. Thus in the old plant, to which reference has already been made, the heat flow even at half-load was 7090 degrees B.T.U. per square foot per hour, while at the full load test at Valley-road the corresponding figure was only 6114 B.T.U. The vacuum in the former case was 27 inches and in the latter 28.89 inches was reached, but it should of course be added that in the latter case the cooling water was supplied 65.7 degrees F. as against 81 degrees F.—"Engineering," July 20, 1928.

THE MEASUREMENT OF LAGGING TEMPERATURES.

The high cost of fuel and the increasing use of high-pressure and highly superheated steam in power-plant operation make it more and more necessary, in the interests of economy and efficiency, to provide satisfactory insulating coverings for steam pipes and boiler drums. According to a recent authority, one superficial foot of bare piping conveying steam at a pressure of 100 pounds per square inch will waste 410 pounds of coal in a normal working year of 3000 hours—or, in other words, two extra tons of coal are required annually for every 10 square feet of uninsulated heat-transmitting surface.

As will be seen from Table 1, the amount of heat lost from an uncovered steam pipe increases as the temperature of the steam is raised. The heat is dissipated by radiation and also, to a considerably reduced extent, by convection and conduction. It is greater when steam is flowing through the pipe than when it is quiescent. The rate of dissipation by radiation depends upon the difference between the temperature of the steam and that of the

outside air, and also on the nature and condition of the hot surface, less heat being lost from a polished exterior than from one which is rough.

By surrounding a pipe with the correct thickness of an efficient lagging, the heat loss may be considerably reduced. Tests have shown that, under practical conditions, the application of an efficient lagging will reduce the amount of heat lost from a bare steam pipe by between 75 per cent and 80 per cent. It does not follow, however, that the heat loss from a bare pipe is always greater than that from a covered pipe. This fact was probably first demonstrated by Péclet when he made various experiments by covering a pipe 12 centimeters in diameter with thicknesses of insulation varying from 1 to 12 centimeters. He found that if the conductivity of the insulation was 0.00012 c.g.s. units, the amount of heat transmitted diminished with increasing thickness; if the conductivity was 0.002, or sixteen times as great, the transmission was approximately constant for all thicknesses, while if the conductivity was 0.004, the amount of heat lost increased with the thickness until at 12 centimenters it was actually greater than that for the bare pipe. These results were subsequently confirmed by various workers,

TABLE I.

External Diameter of Pipe. (Inches.)	Heat loss per hour in B.Th.U. per square feet per degree Fah. temperature difference.			
	At Atmos. Pressure (212 Fah.) (Temp. Diff. 152.)	Steam at 100 lb. (338 Fah.) (Temp. Diff. 278.)	Steam at 200 lb. (388 Fah.) (Temp. Diff. 328.)	Superheated Steam at 500 Fah. (Temp. Diff. 440.)
0·20	3·94	5·40	5·75	6·00
0·50	3·35	4·26	4·50	4·88
1·00	2·92	3·72	3·93	4·30
2·00	2·59	3·37	3·58	3·92
3·00	2·50	3·30	3·48	3·80
6·00	2·36	3·18	3·35	3·70
12·00	2·35	3·14	3·32	3·66

notably by Professor Bottomley, and by Professor A. W. Porter. When the insulating material is either too closely or too loosely packed, the heat loss is increased; in the former case, because of the increased conductivity of the solid material, and in the latter case, because of the greater loss by convection. In general, the amount of money saved by insulating the pipe is less in the case of small pipes than it is for large ones; it is also less for low temperatures than for high temperatures. In any case, however, the sum is sufficiently large to warrant every care being taken to see that the correct thickness of the most suitable form of lagging is used for each particular pipe in the system.

It can be shown that this information may be quickly, easily, and accurately obtained by measuring the surface temperature of the lagging by means of a suitable pyrometer.

The following is the mathematical analysis of the problem:—

Let K = the thermal conductivity of the lagging.

t = the temperature of the steam, that is of the metal pipe considered as a perfect conductor, and therefore of the inner surface of the lagging.

T^1 = the temperature of the outside surface of the lagging.
T = the temperature of the surrounding air.
R_0 = the outside radius of the pipe = the inside radius of the lagging.
R_1 = the outside radius of the lagging.
$d = 2R_0$ = the diameter of the pipe.
$l = R_1 - R_0$ = the thickness of the lagging.
θ = the temperature of the lagging at radius r (which is assumed to remain constant for uniform conditions.)

Then the heat flow through a cylinder of radius r per unit length, per unit time, is

$$-2\pi Kr\frac{d\theta}{dr}$$

This must be a constant for all values of r, as there can be no continuous accumulation of heat at any point, and it is equal to the total heat loss per unit length of pipe.

$$\therefore -2\pi K r\frac{d\theta}{dr} = a$$

Integrating, we have

$$K\theta = -\frac{a}{2\pi}\log r + b$$

$$\therefore K(t - T^1) = \frac{a}{2\pi}\log\frac{R_1}{R_0},$$

or

$$a = K(t - T^1)\,2\pi / \log\frac{R_1}{R_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad 1$$

Knowing the value of K for a particular lagging, it is possible to calculate a, or the total heat loss per unit length. Also experiment shows that the heat loss at the surface of the lagging is given by the formula:—

$$a = 2\pi R_1 C (T^1 - T)^{\frac{5}{4}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad 2$$

where C is a constant and the index $\frac{5}{4}$ is that usually agreed to by the best authorities who have studied the question of heat loss by convection. If T^1—T is measured by means of a pyrometer, and if t is known, the conductivity of the lagging may be found from the equation

$$\frac{K}{C} = R_1 (T^1 - T)^{\frac{1}{4}} \log\frac{R_1}{R_0} / (t - T^1) \quad . \quad . \quad . \quad 3$$

That is to say, the conductivity of any lagging may be found from a single test of any uniform thickness of that lagging on any size of pipe.

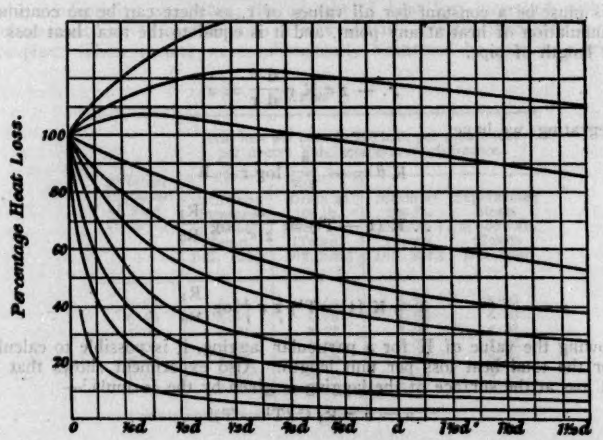
In practice, it is generally permissible to use 1 instead of $\frac{5}{4}$ as the index of $(T^1 - T)$, thus obtaining the simplified formula

$$\frac{K}{C} = R_1 (T^1 - T) \log\frac{R_1}{R_0} / (t - T^1) \quad . \quad . \quad . \quad . \quad 4$$

In industrial practice it is not generally necessary to calculate the actual heat loss, but it is necessary to know the best practical thickness of lagging to use on any given pipe and also the relative merits of different laggings. This latter information can, of course, be obtained by an actual calculation of K in equation (4). If, however, a test is made on different adjacent lengths of the same pipe covered, respectively, with the same thickness of different laggings, these laggings can be arranged in order of merit conversely with the values obtained for their surface temperatures.

The two sets of curves which are reproduced in Figures 1 and 2 will be found helpful when estimating the value of a given lagging under different conditions, that is to say, when used in different thicknesses and on pipes of

Fig. 1. VARIOUS THICKNESSES OF LAGGING ON A PIPE OF DIAMETER *d*.



various diameters. In both sets of curves, the vertical ordinates represent the heat loss from the lagged pipe as a percentage of the heat loss from the same pipe when unlagged. This is the figure of practical interest to engineers. The curves in Figure 1 correspond to different qualities of lagging applied in various thicknesses to a pipe of fixed diameter. The horizontal ordinates give l, the thickness of the lagging, as a fraction of d, the diameter of the pipe.

Using the simplified formula, suppose the value $(T^1 - T)$ is measured by a pyrometer for a 1-inch covering of lagging on a 6-inch pipe and that $(T^1 - T) = 60$ degrees C. and, say, $t - T = 200$ degrees C. Then for the unlagged pipe, the heat loss is proportional to 3 $(t - T)$ and for the lagged pipe it is proportional to 4 $(T^1 - T)$ since $R_0 = 3$ and $R_1 = 4$.

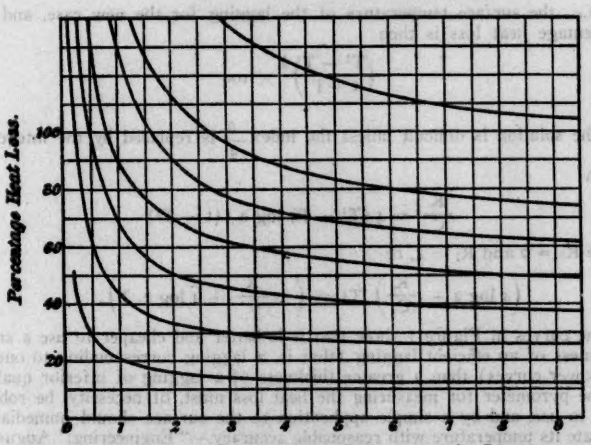
That is, the given lagging saves 60 per cent of the heat lost through the unlagged pipe, and 40 per cent is still lost.

Now refer to the curves in Figure 1.

The thickness of the lagging is $\frac{d}{6}$ and the heat loss is 40 per cent. Select

the curve passing through this point in the diagram. Then the same lagging will give results on this same curve if applied in different thickness to the same diameter pipe. It will be noted at once that considerable saving is effected by increasing the thickness of the lagging to about 2 inches, but beyond this limit the further saving becomes small, and it is certainly extravagant to increase the lagging beyond 3 inches. Other curves can be interpolated by eye between those drawn, as may be required.

Fig. 2. CURVES FOR A THICKNESS l-1" OF LAGGING ON A PIPE OF ANY DIAMETER.



In Figure 2 is given a second set of curves for a fixed thickness 1 equivalent to 1 inch of lagging on any pipe diameter, that is to say, the horizontal ordinates are the diameters of the pipe in inches. Here, again, having found the percentage heat loss as above on a 6-inch pipe, these curves give the percentage heat loss on pipes of other diameters when covered with 1-inch thickness of the same lagging. By means of Figures 1 and 2 it is possible to find a solution for any particular problem.

Suppose, for example, the value of $T^1 - T$ is measured by a pyrometer on a pipe 8 inches in diameter covered with 1-inch of lagging, and it is required to know the corresponding value for a 4-inch pipe covered with 2 inches of lagging. Figure 2 gives a curve corresponding to the reading in question for the 1-inch thickness of lagging on an 8-inch pipe. Following along this curve a reading is found corresponding to 1 inch of lagging on a 4-inch pipe.

In Figure 1 a curve is found for a 4-inch pipe with a lagging $\frac{d}{4}$ in thickness. Follow this curve until the heat loss is found for a 4-inch pipe covered with a lagging $\frac{d}{2}$ or 2 inches thick.

Alternatively, a more accurate calculation may be made as follows:—

For the given data, the value of $\frac{K}{C}$ is found from equation (3) giving

$$\frac{K}{C} = 5\,(T^1 - T)^{\frac{4}{5}} \log \frac{5}{4} \,/\, (t - T^1).$$

Then in the second case, as $\frac{K}{C}$ is now known, we can calculate the new T^1, *i.e.*, the surface temperature of the lagging for the new case, and the percentage heat loss is then

$$\left(\frac{T^1 - T}{t - T}\right)^{\frac{4}{5}} \times 100$$

The solution is difficult unless the index $\frac{5}{4}$ is replaced by the integer 1, when

$$\frac{K}{C} = 4\,(T^1 - T) \log 2 \,/\, (t - T^1)$$

since $R_0 = 2$ and $R_1 = 4$, or

$$\left(4 \log 2 + \frac{K}{C}\right) T^1 = \left(t\,\frac{K}{C} + 4 \log 2.\, T\right).$$

The curves in Figure 1 show that it is better and cheaper to use a small thickness of an efficient lagging (that is, a lagging corresponding to one of the lower curves) than a greater thickness of a lagging of inferior quality.

The pyrometer for measuring the heat loss must, of necessity, be robust, easy to use, and by a simple application to the surface should immediately indicate its temperature with reasonable accuracy.—"Engineering," August 3, 1928.

MACHINERY FOR FAST LINERS.

A year or two ago the rapid development in the design and efficiency of the internal-combustion engine suggested that the sphere of its application could be extended with advantage. Since then actual experience has been obtained in several large ships of more than moderate power with a varying degree of success. The surprisingly low fuel consumption of the oil engine as compared with that of the oil-fired Scotch-boilered turbined ship warranted the adoption of more costly machinery. But if the development in the design of oil engines has been steady, much more dramatic has been that of the high-pressure steam turbine arrangement, which seems to have claimed as its own the powering of ships of large size requiring great power. In the latest of the Canadian Pacific vessels a guarantee of under 0.7 pound of oil per shaft horsepower for all purposes has been accepted, and when the weight of an installation of the type of machinery to be installed is compared with that of internal-combustion engines developing the same power any advantage due to the greater fuel economy of the latter tends to disappear. Not only so, but internal-combustion engines cost very much more initially, and perhaps the risk of heavy upkeep costs is much greater than with turbine machinery.—"Shipbuilding and Shipping Record," July 12, 1928.

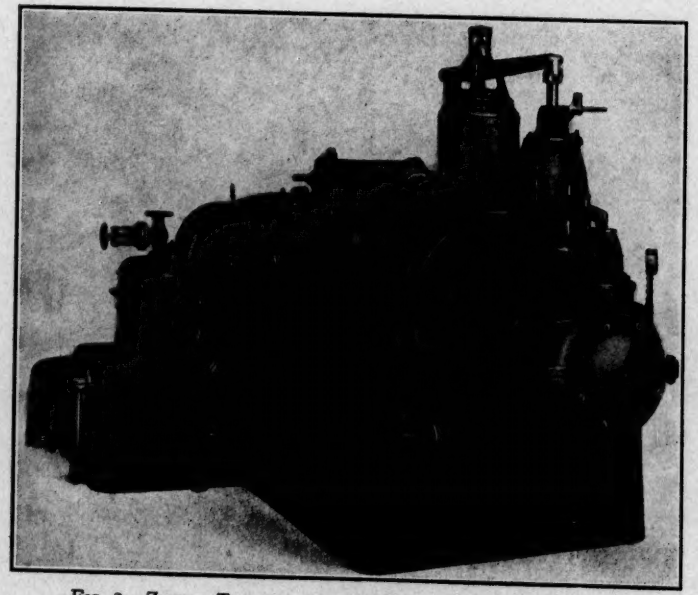


FIG. 2.—ZOELLY TURBINE FOR 1420 POUNDS PER SQUARE INCH.

TURBINES FOR VERY HIGH STEAM PRESSURES.

Although the theoretical advantages to be derived from the use of steam of a very high pressure and temperature have been discussed very fully, the realization of these advantages in practice has incurred a deal of research work, especially in regard to the behavior of materials under high temperature and pressure. However, sufficient progress has been achieved to show the practicability of using very high-pressure steam turbines with very substantial advantages in heat economy. Thus with a high-pressure turbine functioning ahead of the normal turbo-generator set, the former can be mechanically coupled to the latter by speed-reducing gear, or operated as a separate unit driving an electric generator. It will, of course, frequently be found desirable to reheat the steam before passing it to the rear turbine.

The steam turbines for high pressures and temperatures illustrated herewith are by Escher, Wyss and Co., Zurich, a firm which has given considerable attention to the design of such plant. Pressures of 425 pounds and 570 pounds per square inch have been adopted in practice chiefly because the normal type of boiler can be employed; but pressures of 570 pounds to 850 pounds per square inch no longer present any serious difficulties, provided due attention is given to the effect of the concomitant high temperatures upon the structure and allowance made for the expansion of the

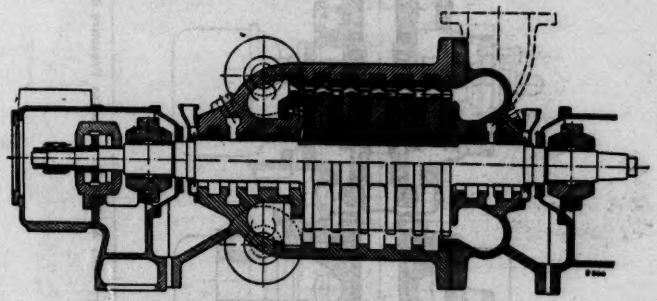


Fig. 1.—Zoelly Turbine for a Steam Pressure of 780 lb. per sq. in.

various elements. It is to be noted that with pressures of from 570 pounds to 850 pounds at the stop-valve of the main turbine, together with the highest possible temperature of about 840 degrees F., it is possible to employ continuous expansion to the condenser with good efficiency, without the necessity for intermediate reheating. Figure 1 shows a Zoelly turbine operating under the conditions described, and placed ahead of a condensing turbine. The admission pressure is 780 pounds per square inch, and the steam temperature 840 degrees F. Figure 2 shows a Zoelly turbine erected in the power station of the Siemens-Schuckert Works in Berlin. Steam is supplied by a Benson boiler at an admission pressure of 1420 pounds per square inch. The temperature is 750 degrees F.; the back pressure ahead of the intermediate reheater is 213 pounds per square inch; and the steam passing is 10 tons per hour. The experience obtained with this turbine has led to modifications in the design of fittings, joints, blading, etc., and has also been of service in the study of certain dynamical phenomena due to the great density of the steam. The casing of this turbine is made of one single

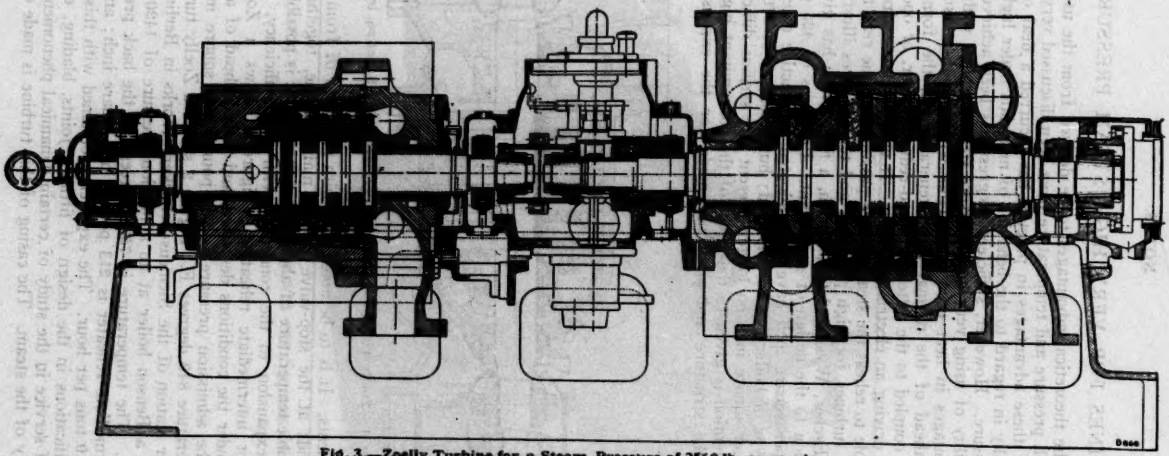


Fig. 3.—Zoelly Turbine for a Steam Pressure of 2560 lb. per sq. in.

piece of forged steel, while the rotor, with the exception of the blading, is also in one piece.

In view of the favorable results obtained with this turbine the Siemens-Schuckert Works has placed an order with Escher, Wyss & Co. for a turbine in which the exceptionally high admission pressure of 2560 pounds per square inch is employed, the steam temperature being 790 degrees F. The turbine, a section of which is given in Figure 3, is designed for two intermediate tappings, and back pressure in two casings with intermediate reheating between the two casings. Steam is supplied by a special form of Benson boiler. The normal quantity of steam used per hour is 20 tons, the approximate output being 3500 brake horsepower. It is expected that the turbine will be put into service at the end of the year, and the results will be awaited with interest.—"Mechanical World and Engineering Record," August 24, 1928.

BOOK REVIEW.

THE WAR IN THE AIR, VOLUME II, BY H. A. JONES. 17 s. 6 d. 9 × 6; 508 pages, with 13 maps. London: The Oxford University Press.

This is the second volume of the history of the part played in the war by the British air forces. In volume I, Sir Walter Raleigh dealt with the beginnings of British aviation and its achievements during the autumn and winter of 1914. The present volume was undertaken by the author along the same lines as those of the first volume. The air story is built up on a framework of the naval and military operations, but only such of them in which aviation played a part are included.

The second volume opens with the air operations of the Dardenelles campaign and after the evacuation of Gallipoli in January, 1916, the scene shifts to the Western Front from the winter of 1914-15 to the end of the Somme battles in November, 1916. The final chapters deal with the Royal Naval Air Service in home waters; the naval air operations from Dunkirk in 1915 and 1916; and the bombing operations from Luxeuil in the latter part of 1916.

While the military or naval operations on the ground as opposed to those from the air are merely given in outline form, the air preparations and actual encounters are presented in minute detail. At times, in fact, the story reads as if it were the log of an individual squadron with each pilot's name, his time of take-off and landing, and the number and weight of each bomb dropped, chronicled in full. The author's research for these details must have been tremendous, but there appears no great attempt to match the many exploits recounted with detailed German records. Numerous items of enemy history are, however, presented and the story of the gallant Oswald Boelcke, his brilliant leadership, and his last success, is one of the most interesting in the whole book. Of the

air operations of the battle of Jutland, the author states, "Its interest to the student of air power lies, not of what aircraft did, but of what opportunities of perhaps vital importance were open for their use."

An error is made in the footnote on page 152 in crediting the late Captain Sir John Alcock as the first to navigate an airplane across the Atlantic. Commander A. C. Read, U. S. Navy, preceded Alcock by nearly a month, albeit Commander Read's flight was made with a stop at the Azores. A typographical error occurs in the formula on page 463. The approximate time of fall of a bomb is one-fourth the *square root* of the altitude, in feet, of release.

This book is well printed, has many notes and an excellent index for both volumes I and II. It is a first rate piece of work.

REVIEWED BY LIEUTENANT COMMANDER J. E. OSTRANDER, JR., U. S. N.

ASSOCIATION NOTES.

RESIGNATION OF PRESIDENT.

Because of detachment from duty in Washington, as Engineer-in-Chief and Chief of the Bureau of Engineering, Navy Department, Rear Admiral John Halligan, U. S. Navy, resigned as President of the Society effective 1 September, 1928. The Council has appointed Captain E. L. Bennett, U. S. Navy, as President to fill the unexpired term of Admiral Halligan.

CHANGE IN COUNCIL.

The appointment of Captain E. L. Bennett, U. S. Navy, as President of the Society, created a vacancy in the Council, of which he was a member. The Council has appointed Captain I. E. Bass, U. S. Navy, to fill this vacancy.

ANNUAL MEETING AND NOMINATION OF OFFICERS.

The Annual Meeting of the Society to nominate officers for the coming year was held at Washington, D. C., on Tuesday, October 2, 1928. The following were nominated:

For President:

Rear Admiral Harry E. Yarnall, U. S. N.

For Secretary-Treasurer:

Commander H. T. Smith, U. S. N.

For Member of Council:

Rear Admiral C. W. Dyson, U. S. N., Retired.
Captain E. L. Bennett, U. S. N.
Captain R. B. Adams, U. S. C. G.
Captain I. E. Bass, U. S. N.
Captain O. L. Cox, U. S. N.
Commander H. S. Howard (CC), U. S. N.
Mr. B. P. Lamberton.
Mr. H. M. Southgate.
Mr. J. F. Metten.

ANNUAL BANQUET.

At the Annual Meeting it was decided that a banquet of the Society should be held in 1929. The President will appoint a committee and full details will be given at as early a date as practicable.

PROPOSED AMENDMENTS TO BY-LAWS.

Amendments to the By-Laws of the Society were proposed at the Annual Meeting, the effect of which is (1) to provide for dropping of members in arrears for dues for two years; (2) to permit Civil Members to vote; (3) to provide a committee to present nominations to offices at the Annual Meeting.

MEMBERSHIP.

The following members have joined the Society since the publication of the August, 1928, JOURNAL:

NAVAL.

Nasi, George A., Electrician, U. S. Navy.

CIVIL.

McKee, J. E., Room 300, 30 Church St., New York, N. Y.

Riis, S. M., Standard-Noble Co., Al. Jerozolimskie 57, Warsaw, Poland.

ASSOCIATE.

Glen, D. S., 355 Mountain Way, Rutherford, N. J.

Lu, W. S., Electrical Engineer, Kiangnan Dock, Shanghai, China.